

TRANSACTIONS

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

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AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

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Meets: First Tuesday in Month

CONTENTS

CHAPTER	PAGE
678 THE 30TH ANNUAL MEETING.....	1
679 CODE FOR TESTING LOW-PRESSURE STEAM HEATING BOILERS..	9
680 ECONOMICAL UTILIZATION OF HEAT FROM CENTRAL STATION PLANTS, BY N. W. CALVERT AND J. E. SEITER.....	21
681 AN IMPROVED METHOD OF DETERMINING THE HEAT TRANSFER THROUGH WALL, FLOOR AND ROOF STATIONS, BY R. F. NORRIS, H. H. GERMOND AND C. M. TUTTLE.....	41
682 DETERMINING THE EFFICIENCY OF AIR CLEANERS, BY A. M. GOODLOE.....	47
683 AIR HANDLING AND HUMIDITY PROBLEMS IN A WISCONSIN PAPER MILL, BY ARTHUR T. NORTH.....	55
684 HEAT EMISSION FROM HEATING SURFACES OF FURNACE, BY A. P. KRATZ.....	59
685 MEASURING HEAT TRANSMISSION IN BUILDING STRUCTURES AND A HEAT TRANSMISSION METER, BY P. NICHOLLS.....	65
686 AIR LEAKAGE THROUGH OPENINGS IN BUILDINGS BY, F. C. HOUGHTEN AND C. C. SCHRADER.....	105
687 THE PRODUCTION AND MEASUREMENT OF AIR DUSTINESS, BY MARGARET INGELS.....	121
688 CRITICAL VELOCITY OF STEAM AND CONDENSATE MIXTURES IN HORIZONTAL, VERTICAL AND INCLINED PIPES, BY F. C. HOUGHTEN, LOUIS EBIN AND R. L. LINCOLN.....	139
689 SIMULTANEOUS FLOW OF WATER AND AIR IN PIPES, BY L. S. O'BANNON.....	157
690 AIR MOTION-HIGH TEMPERATURES AND VARIOUS HUMIDITIES- REACTIONS ON HUMAN BEINGS, BY W. J. McCONNELL, F. C. HOUGHTEN AND C. P. YAGLOGLOU.....	167
691 COOLING EFFECTS ON HUMAN BEINGS PRODUCED BY VARIOUS AIR VELOCITIES, BY F. C. HOUGHTEN AND C. P. YAGLOGLOU.	193
692 THE PLACE OF ELECTRICITY IN THE GENERAL HEATING FIELD, BY LEE P. HYNES.....	212
693 PROBLEMS IN THE VENTILATION OF DEPARTMENT STORES, BY A. M. FELDMAN.....	221
694 THE STATUS OF DOMESTIC OIL HEATING, BY A. H. BALLARD...	227
695 SEMI-ANNUAL MEETING, 1924.....	235
696 USING BY-PRODUCTS IN FLOUR MILL HEATING AND HU- MIDIFYING, BY E. K. CAMPBELL.....	243

697	DETERMINING DRY RETURN PROPORTIONS, BY R. V. FROST....	253
698	OZONE AND ITS USE IN VENTILATION, BY FRANK E. HARTMAN.	259
699	PERFORMANCE OF A WARM AIR FURNACE WITH ANTHRACITE AND BITUMINOUS COAL, BY A. P. KRATZ.....	277
700	SELECTING WALL STACKS SCIENTIFICALLY FOR GRAVITY WARM AIR HEATING SYSTEMS, BY V. S. DAY.....	284
701	PRACTICAL APPLICATION OF THE HEAT FLOW METER, BY P. NICHOLLS.....	289
702	VALUE OF THE KATA THERMOMETER IN EFFECTIVE TEMPERA- TURE STUDIES, BY MARGARET INGELS.....	301
703	CORRELATION OF SKIN TEMPERATURES AND PHYSIOLOGICAL REACTIONS, BY W. J. McCONNELL AND C. P. YAGLOGLOU.	305
704	AIR LEAKAGE AROUND WINDOW OPENINGS, BY C. C. SCHRADER.	313
705	FLOW OF STEAM AND CONDENSATION AS AFFECTED BY HIGH PRESSURES, HORIZONTAL OFFSETS AND VALVES, BY LOUIS EBIN AND R. L. LINCOLN.....	323
706	EFFECTIVE TEMPERATURE APPLIED TO INDUSTRIAL VENTILA- TION PROBLEMS, BY C. P. YAGLOGLOU AND W. E. MILLER.	339
707	HEAT GIVEN UP BY THE HUMAN BODY AND ITS EFFECT ON HEATING AND VENTILATION PROBLEMS, BY C. P. YAGLOGLOU.	365
708	MODERN TREND IN THE SCIENCE OF VENTILATION, BY PERRY WEST.....	377
709	SOME COMMENTS ON PRESENT DAY HEATING AND VENTILATION PRACTICE, BY W. S. TIMMIS.....	395

TRANSACTIONS

of

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

No. 678

THE THIRTIETH ANNUAL MEETING, 1924

THREE DECADES of growth and progress were celebrated at the thirtieth Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, held at the Hotel Pennsylvania, New York City, January 22 to 25. The session, which was the longest ever held, was featured by two events of outstanding importance—the approval of the preliminary draft of the Code of Minimum Requirements for the Heating and Ventilation of Buildings and the decision to operate the Research Laboratory under a new set of regulations. The revised Code for Testing Low-Pressure Heating Boilers was also adopted, and the members unanimously approved the comments of the Committee appointed to consider the Report of the New York State Ventilation Commission.

The business session opened on Tuesday morning, January 22, with Pres. H. P. Gant presiding, who outlined briefly the work of the administration during the year. This was followed by reports of officers and committees, and the announcements of the election of officers as follows:

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Homer Addams, 47 West 42nd St., New York, N. Y.

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Section VI.	Heating Boiler Capacity.....	J. F. McIntire
Section VII.	Warm Air Furnace Heating.....	J. D. Hoffman
Section VIII.	Design of Chimneys and Flues.....	J. R. McColl
Section IX.	Pipe Sizes for Steam Heating.....	J. A. Donnelly
Section X.	Pipe Sizes for Hot-Water Heating.....	W. S. Timmis
Section XI.	Air Ducts for Ventilation.....	C. A. Booth
Section XII.	Air Washers and Humidifiers.....	W. H. Carrier
Section XIII.	Pumps for Heating Systems.....	Perry West
Section XIV.	Standard Symbols for Drawings.....	J. H. Walker

More than 300 members and guests attended the meeting, including 50 women. A reception and informal dance were held in the small ballroom of the Hotel Pennsylvania on Tuesday evening. Helen R. Innes, Mrs. C. W. Obert, and Mrs. A. A. Kiewitz received the women and the entire New York Chapter Committee greeted the men.

Tea was served for the women guests, in the main dining room of the hotel, on Wednesday afternoon.

A dinner-dance, held in the grand ballroom of the hotel on Thursday evening, concluded the social program of the Meeting. H. P. Gant, retiring president, was presented with a watch-fob in the form of the Society emblem, set with diamonds, in token of the appreciation and esteem of the Society. The music was so good and everyone present enjoyed himself so much that dancing was continued until long after midnight.

The New York Chapter was congratulated upon its efficient handling of all phases of the meeting and the guests extended to its members a hearty vote of thanks and appreciation.

REPORT OF COMMITTEE TO CONSIDER THE REPORT OF NEW YORK STATE COMMISSION ON VENTILATION

THE report of the New York State Commission on Ventilation consists of two parts. The first being a study of the physiological significance of the various factors in ventilation, and the second, a study of the practical results achieved by the use of various methods of schoolroom ventilation.

From the point of view of the engineer the most important contribution which this report makes to the science of heating and ventilation is contained in the first part of the report.

It is shown for instance in Chapter IX that men work much less effectively in a hot humid room than in a relatively cool and dry atmosphere. The report shows that men accomplish 28 per cent less physical work in a temperature of 86 deg. fahr. with 80 per cent relative humidity than in a room 68 deg. fahr. with 50 per cent relative humidity. Other tests establish the fact beyond question that high temperatures particularly when combined with high humidity do reduce the capacity for physical work. On the other hand the interesting fact is disclosed in Chapter X that purely mental work is only slightly effected by a rise in temperature from 68 deg. to 75 deg. fahr. It is probably the capacity for physical work in such a test as typewriting, for example, that is effected and not the mentality of the worker. The following is the conclusion arrived at after a series of experiments on vitiated air:

Appetite and Growth. Finally, we found a marked influence exerted by stale air upon the appetite for food as determined by serving standard lunches to parallel groups of subjects, in stale and fresh air, respectively, but with the same temperature and humidity. In the four different series of experiments which were successfully completed on this basis without the intrusion of interfering factors, the excess of food consumed under

fresh air conditions was respectively, 4.4, 6.8, 8.6, and 13.6 per cent. Since the probable errors involved in these experiments were relatively very slight it seems evident that the chemical constituents of vitiated air may not only diminish the tendency to do physical work but also the appetite for food.

The scope of the experiments and the number of subjects observed made the work of the Commission unique and add greatly to its value.

The conclusions in regard to CO_2 in the report are also valuable.

Chapter XII deals with the effect of high temperature and humidity on the mucous membrane of the nose and throat and shows that heat and particularly moist heat produces a red and swollen condition of the mucous membrane and that a large percentage of people working in such conditions develop atrophic rhinitis.

Part II of the report is full of interesting and stimulating observations and suggestions. The numerous "conclusions" seem in some cases prematurely arrived at, but there is much excellent material which may well be studied by the ventilating engineer.

Chapter XXIV is of particular interest as it attempts to show the distribution of air to be expected by locating the inlets and outlets for air in different parts of the room.

The greatest disappointment in this gigantic piece of work by the New York State Commission on Ventilation is that practically they have brought us no nearer the knowledge of what the fundamentals of ventilation should be. Here was a great opportunity to settle disputed theories of long standing. It is true that some side lights have been given on a few of the factors of ventilating, but these, for the most part, confirm existing beliefs and do not settle, in any way, the various basic requirements in ventilation. If this Commission, composed mostly of well known doctors and scientists, could have attacked the problem from the standpoint of determining, once for all, the exact fundamentals in ventilation, the methods of producing these atmospheric conditions would most surely follow, for ventilating engineers are able to produce any predetermined atmospheric conditions if they know what these should be. It must be remembered, however, that these tests were made prior to 1917. Since that time great progress has been made toward establishing factors of ventilation. These standards in the last two or three years in particular, are being crystallized into pretty definite shape, as the results of the work of other scientific bodies, principally the work in the Research Laboratories in Pittsburgh, conducted, jointly, by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, United States Bureau of Mines and United States Public Health Service. As compared with this work in Pittsburgh, any conclusions on what constitutes real ventilation as brought out by the tests of the New York State Commission on Ventilation, are indefinite and incomplete.

It is to be regretted that the Commission did not have time or means to carry out the program of investigation as originally contemplated. As the report stands the conclusions cover only a portion of the ventilation study at first proposed by the Commission.

Though extensive and painstaking were the tests of systems by the Commission in New York, they are only, after all, a comparison between window ventilation and the mechanical ventilating systems of some New York Schools. Apparently the investigators have assumed, and without doubt, honestly, that the mechanical ventilating systems in the school buildings tested were representative of such systems. This is far from true as there are many schools elsewhere in the country which have gone to far greater refinement in air conditioning and air distribution than those in which the tests were conducted.

The attempt to make, through the medium of questionably designed systems, comparisons of the broad relative merits of window ventilation and fan ventilation is misleading and unwarranted.

Much is said in the report about various factors, particularly high temperature and drafts, in mechanically ventilated rooms, the impression being conveyed that these are inherent and incurable defects in mechanical systems. Nothing is more erroneous in a well designed and operated system, under automatic control. Partial hand control is recommended. This amounts to a charge that automatic control systems cannot be a complete success. Furthermore, it puts an unwarranted additional burden on the teacher. This burden is always present in the window ventilation. If variations in temperature, air movement, and even humidity are desirable in schoolroom ventilation, these can be taken care of automatically in a mechanical system. But

first the physiologists must give the curves for these variations. The conclusions given on page 468 of the report as to the proper design of a mechanical system do not apply generally. The tests on humidification and recirculation outlined in Chapters XXV and XXVI, although representing an immense amount of work, are not extensive enough to give data upon which definite conclusions can be based.

There are many questions which can be raised as to the reliability of certain data given in the report, as the conclusions reached were dependent upon primary sense of the observer or teacher. The anemometer at its best is not an accurate instrument, and when placed on the vent screen gives anything but a true reading for air volumes.

A ventilation standard is the all important subject to the heating and ventilating engineer.

Two principles are at the very foundation of this ventilation progress structure:

1. No specific apparatus or method should be set forth in defining a standard.
2. The atmospheric conditions desirable should be the all important and only consideration.

Therefore:

Ventilation perfection is attained when the atmospheric condition in every part of a room occupied by human beings is continually maintained with normal amount of oxygen and free from dust, bacteria, odors and poisons with suitable air movement and at the temperature and humidity quality shown within the comfort zone as determined by the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

The apparatus or method that most nearly causes atmospheric air to meet this ventilation perfection or ideal is the most satisfactory.

PROGRAM ANNUAL MEETING 1924

First Session

Tuesday, January 22, 10 a.m.

Hotel Pennsylvania, New York, N. Y.

BUSINESS SESSION

Announcement of Quorum
Appointment of Tellers of Annual Election
Address of President
Report of Council
Report of Secretary
Report of Treasurer
Report of Committee

- a. Executive
- b. Finance
- c. Publication
- d. Membership

Report of Tellers of Election
New Business

Second Session

Tuesday, January 22, 2 p.m.

Hotel Pennsylvania, New York, N. Y.

PROFESSIONAL SESSION

Paper:
Problems in Ventilation of Department Stores, by A. M. Feldman
Paper:
Determining the Efficiency of Air Cleaners, by A. M. Goodloe
Paper:
Air Handling and Humidity Problems in Wisconsin Paper Mill, by Arthur T. North
Paper:
The Place of Electricity in the General Heating Field, by L. P. Hynes

Third Session

Wednesday, January 23, 10 a.m.
Hotel Pennsylvania, New York, N. Y.

PROFESSIONAL SESSION

- Paper: An Improved Method of Determining the Heat Transfer through Wall, Floor and Roof Sections, by Norris, Germond and Tuttle
- Paper: Economical Utilization of Heat from Central Station Plants, by Calvert and Seiter
- Paper: Heat Emission from Heating Surfaces of Furnace, by A. P. Kratz
- Paper: The Status of Domestic Oil Heating, by A. H. Ballard

Fourth Session

Wednesday, January 23, 2 p.m.
Hotel Pennsylvania, New York, N. Y.

RESEARCH SESSION

Report of Research Committee

- Address: Research and Industrial Progress, by F. Paul Anderson, Director
- Paper: Measuring Heat Transmission in Building Structures and a Heat Transmission Meter, by P. Nicholls
- Paper: Critical Velocity of Steam and Condensate Mixtures in Horizontal, Vertical and Inclined Pipes, by Louis Ebin, F. C. Houghten and R. L. Lincoln
- Paper: Checking Up the Kutter Formula and Its Application to the Flow of Liquids and Gases through Small Pipes, by L. S. O'Bannon

Fifth Session

Wednesday, January 23, 8 p.m.
Hotel Pennsylvania, New York, N. Y.

RESEARCH SESSION

- Paper: Air Motion, High Temperature and Various Humidities Affecting Physiological Reactions of Human Beings, by Dr. W. J. McConnell, C. P. Yagloglou and F. C. Houghten
- Paper: Cooling Effect on Human Beings Produced by Various Air Velocities, by F. C. Houghten and C. P. Yagloglou
- Paper: Air Leakage through the Openings in Buildings, by F. C. Houghten and C. C. Schrader
- Paper: The Production and Measurement of Air Dustiness, by Margaret Ingels

Sixth Session

Thursday, January 24, 10 a.m.
Hotel Pennsylvania, New York, N. Y.

CODE SESSION

Report of Proposed Code of Heating and Ventilating
Discussion conducted by L. A. Harding, *Chairman*

Seventh Session*Thursday, January 24, 2 p.m.*

Hotel Pennsylvania, New York, N. Y.

CODE SESSION

Report on Proposed Code of Heating and Ventilating
Discussion conducted by L. A. Harding, *Chairman*

Eighth Session*Friday, January 25, 10 a.m.*

Hotel Pennsylvania, New York, N. Y.

VENTILATING SESSION

Report of Committee to Consider the Report of the New York State Commission on
Ventilation, by C. L. Riley, *Chairman*

Ninth Session*Friday, January 25, 2 p.m.*

Hotel Pennsylvania, New York, N. Y.

CLOSING SESSION

Unfinished business

Installation of officers

No. 679

CODE FOR TESTING LOW-PRESSURE STEAM-HEATING BOILERS

Revision of 1923

YOUR Committee on Code for Testing Low-Pressure Steam-Heating Boilers submits herewith for the consideration of the Annual Meeting a proposed revision of the Testing Code which when approved, is to be known as the Revision of 1923.

Committee on Code for Testing Low-Pressure
Steam-Heating Boilers

{ JOHN BLIZARD, *Chairman*
HOMER ADDAMS
F. PAUL ANDERSON
L. P. BRECKENRIDGE
P. J. DOUGHERTY
L. A. HARDING
F. B. HOWELL
J. F. MCINTIRE

CODE FOR TESTING LOW-PRESSURE STEAM-HEATING BOILERS

1923

Object of the Code

THE object of the Code for Testing Low-Pressure Steam-Heating Boilers is to provide a standard method for conducting and reporting tests to determine the heat efficiency at various rates of steaming.

Essentials Necessary to Determine Heat Efficiency

The essentials necessary to determine the heat efficiency of a steam-heating boiler are:

- a. The total heat input.

(The total heat input is the total heat value of the fuel charged.)

- b. The total heat recovered at the boiler outlet.

(The total heat recovered at the boiler outlet is the total heat of the steam leaving the boiler less the total heat of the feed water entering the boiler.)

Preparations for Test

The boiler shall be erected, covered and connected to conform to the directions and practice of the manufacturer. The piping shall be connected in such a way that the steam may be carried to a point away from the boiler and it shall be arranged so that the condensation cannot flow back to the boiler.

The moisture in the steam shall be determined by a steam separator, not less than 95 per cent efficient, placed in the steam delivery pipe as close to the boiler as possible. The piping between this separator and the boiler, also the separator itself, shall be thoroughly covered with insulating material. A pipe connected to the bottom of the steam separator shall be provided with a positive seal. The water shall be drained from the separator hourly and weighed immediately.

The steam connections between the boiler outlet and the separator shall be the same in size and arrangement as that to be used when the boiler is installed.

The water shall be fed to the boiler continuously from the feed tank through piping with all necessary valves, and all other water connections to the boiler shall be carefully blanked off. The temperature of the feed water shall be read from a thermometer inserted in a cup projecting well into the feed line near the boiler and filled with a heavy oil. All boiler water connections, including blow-off pipes, must be exposed to view, so that leakages may be observed, and either stopped or measured. The glands of the feed pump shall be carefully packed to prevent leakage.

The boiler shall be connected with a short, direct smoke-pipe to a chimney flue of suitable size, height and construction to give proper draft.

The water spaces of the boiler shall be thoroughly boiled out with a solution of sal soda, potassium hydrate or sodium hydrate and then thoroughly rinsed with clean water.

The heating surface, firebox, ashpit, flues and chimney shall be clean and free from soot, ashes and dust at beginning of test.

Apparatus and Instruments

Apparatus and instruments must be reliable and be arranged in such a way as to insure correct data.

Tanks for measuring the feed water may be calibrated with weighed quantities of water at the temperature to be used during the test, or mounted on accurate weighing scales. The water may be fed to the boiler by gravity, by air pressure or by feed pumps, from feed-water tanks supplied from the measuring tanks by gravity.

Accurate scales of suitable size shall be provided for weighing separator water, fuel and all refuse removed from the grate and ashpit.

Three draft gages shall be provided and so arranged as to determine the pressure difference at the level of inserting the pipe between the outside and the ashpit, between the outside and the firebox, and the outside and the smokehood. Draft measurements shall be made with draft gages reading to 0.01 in.

Accurately calibrated instruments shall be provided for measuring temperatures of gases, water and steam.

An Orsat apparatus shall be used for determining the flue gas composition. If recording carbon dioxide (CO_2) instruments are provided, they shall be checked every hour with the Orsat apparatus.

A Ringelmann chart shall be used for smoke observations.

Weather Bureau reports from the immediate vicinity may be used to determine the barometric pressure. When such reports are not available, a calibrated aneroid barometer or mercury column shall be used for determining the barometric pressure.

A calibrated steam gage or a mercury column shall be used for determining the steam pressure.

A log of the test shall be kept on record sheets similar to those provided by this Code.

Duration of Test

The test shall continue for at least 16 hours if operated at the normal manufacturer's rating; if operated at other ratings it shall continue until as much fuel has been burned as would have been burned in a 16-hour test at normal rating.

Method of Starting and Stopping Test

The New Fire Method of starting and stopping test may be used on any boiler when anthracite coal is used as fuel. All tests using other fuels shall be started and stopped by the Continuous Firing Method.

New Fire Method. A preliminary fire shall be made and the boiler operated under test conditions for at least one hour before starting the test. The preliminary fire shall then be dumped, the ashpit thoroughly cleansed and dried wood placed on the grate and kindled. The test shall be considered started at the time of firing the charge of wood. On this charge of wood, fuel shall be placed. The wood shall be considered as having a heating value of 5000 B.t.u. per lb. The height of water line in gage glass and feed tank shall be noted and recorded at the time the preliminary fire is dumped. The water level in the boiler shall be kept at this level as nearly as possible throughout the test and the water level in the boiler and feed tank must stand at this same height when the test closes. At the end of test the fire shall be dumped. The residual fire when dumped shall be placed in tightly covered cans, weighed and left to cool. After cooling it shall be forwarded for analysis and determination of its heat value and ash content. The total fuel fired shall be taken as the total weight of fuel exclusive of the wood used for kindling, to which shall be added the fuel equivalent of the wood and from which shall be subtracted the fuel equivalent of the residual fire. The weight of the ash content of the residual fire shall be added to the weight of ash and refuse removed from the ashpit and the sum recorded as ash and refuse removed from the ashpit.

Continuous Firing Method. A preliminary fire shall be made and the boiler operated under test conditions for at least one firing period and not less than one hour before starting the test.

The fire shall then be burned low, thoroughly cleansed and the remaining live fuel spread evenly over the grate as the foundation for the first test fuel charge. The thickness of the fuel bed and the extent to which it has been burned through shall be quickly estimated or measured. The height of water line in gage glass and feed tank shall be noted and recorded. The test shall start at the time of making these observations. A weighed charge of fuel shall then be fired. The ashpit shall be thoroughly cleansed immediately and the test allowed to proceed.

A constant water level and rate of steaming shall be maintained throughout the test.

At the end of the test the fire should be burned low and cleansed so as to leave the same amount of live fuel on the grate as at the start. When this condition is reached and the water level in the boiler and feed tank are at the same height as at the start, record the time and this time shall be the time of stopping. The contents of the ashpit shall be removed promptly on stopping and placed in airtight cans, weighed and left to cool. The boiler shall be charged with all fuel charged during test.

Method of Firing

The method and frequency of firing shall be as agreed upon by the manufacturer and purchaser.

Fuel Sampling

During the progress of the test, fair samples at regular intervals shall be taken with a shovel from the fuel charge, stored in a covered vessel in a cool place, and after crushing and quartering, two one-pint glass jars or other airtight vessels shall be filled. The gross sample for slack coal and small sizes of anthracite in which the impurities do not exist in abnormal quantities or in pieces larger than $\frac{3}{4}$ in., should weigh approximately 500 lb. and not less than 1000 lb. for other solid fuels.¹

The small samples shall be preserved for determinations of the proximate analysis, ultimate analysis and calorific value.

The refuse taken from the ashpit and grate shall be reduced by crushing and quartering to a quantity sufficient to fill two one-pint jars or other airtight vessels for determining its combustible content in the laboratory. Care must be taken to crush and quarter the coal, ash, and refuse on a clean floor; to avoid contaminating the sample a metal plate is to be preferred to a concrete floor. Care must be taken to see that the ash and refuse does not burn after removal from the grate or ashpit.

STANDARD FORM

For Reporting Results of Low-Pressure Steam-Heating Boiler Tests

RESULTS

Of a Test on a Low-Pressure Boiler

DATE OF TEST.....
 CONDUCTED AT.....
 DIRECTOR OF TEST..... (signature)
 MANUFACTURER OF BOILER.....
 OWNER OF BOILER.....
 SIZE OF BOILER.....
 TYPE OF BOILER.....

¹ As recommended by the *American Society for Testing Materials*, D21-16, page 756, 1921.

These forms may be obtained on request at the office of the Secretary of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, 29 West 39th Street, New York City, at the nominal cost of per copy.

The Committee on Code for Testing Low-Pressure Heating Boilers of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS is prepared to interpret the meaning of any items on the Code.

It is requested that all tests be filed with the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

GENERAL PARTICULARS OF BOILER AND FUEL

BOILER

Type.....	
Made by.....	
Length of Grate (or Diameter).....	in.
Width of Grate.....	in.
Fuel Capacity (Greatest Possible Volume).....	cu. ft.
Maximum Fuel Depth (Greatest Possible Depth of Fuel).....	in.
Fuel Capacity Normal.....	cu. ft.
Fuel Depth Normal.....	in.
Average Distance from Top of Normal Fuel Charge to Crown Sheet.....	in.
Total Furnace Volume, Grate to Crown Sheet and Bridge Wall.....	cu. ft.
Total Combustion Space Beyond Bridge Wall.....	cu. ft.
Water Capacity (To Water Line).....	lb.
Heat of Water Line.....	in.
Steam Connections Used	<div> <div>Number.....</div> <div>Size.....</div> </div>
Kind of Insulation.....	
Thickness of Insulation.....	in.
Detailed Description of Boiler.....	

SMOKE PIPE AND CHIMNEY

Area of Smoke Pipe.....	sq. in.
Length of Smoke Pipe (Boiler to Chimney).....	in.
Number and Kind of Bends in Smoke Pipe.....	
Chimney, height above Grate.....	ft.
Chimney, Area at Bottom.....	sq. ft.
Chimney, Area at Top.....	sq. ft.

FUEL

Name.....	
Size.....	

PROXIMATE ANALYSIS

		As fired	Moisture free
Moisture.....	per cent
Volatile Matter.....	per cent
Fixed Carbon.....	per cent
Ash.....	per cent

ULTIMATE ANALYSIS

		As fired	Moisture free
Carbon.....	per cent
Hydrogen.....	per cent
Oxygen.....	per cent
Nitrogen.....	per cent
Sulphur.....	per cent
Ash.....	per cent

HEAT VALUE (GROSS)

B.t.u. per lb. as fired.....	
B.t.u. per lb. moisture free	
B.t.u. per lb. moisture and ash free.....	

CHARACTER OF FUEL

(State whether coking or free-burning, clinker troubles, etc.)

METHOD OF FIRING

PRINCIPAL RESULTS OF TEST

Heat recovered at the boiler outlet per hour.....	1000 B.t.u.
Maker's rating (sq. ft. radiation \times 240).....	1000 B.t.u. per hr.
Percentage of maker's rating developed.....	per cent
Mean interval between charging fuel.....	hours
Mean interval between attention of any kind to the fire, including charging.....	hours
Overall thermal efficiency.....	per cent

DETAILED RESULTS OF TEST

(For full particulars of boiler and fuel see "general particulars" ante)

GENERAL INFORMATION

1. Date of Test.....
2. Number of Test.....
3. Location of Boiler.....
4. Maker of Boiler and Type.....
5. Owner of Plant.....
6. Test Conducted by.....
7. Duration of Test..... hr.
8. Manufacturer's Rating of Boiler..... sq. ft. radiation¹
9. Grate Area²..... sq. ft.
10. Barometric Pressure..... in. of mercury

¹ One sq. ft. radiation to be assumed equal to 240 B.t.u. per hr.

² If the grate has an unusual shape, method of computing area must be stated under "Remarks."

FUEL

11. Heat value, as fired.....B.t.u. per lb.
12. Number of Times Fuel Charge during Test.....lb.
13. Intervals between Charging, hrs. Longest.....Shortest.....Average.....
14. Intervals between Attention of any Kind to the Fire, including firing, hr.
 Longest.....Shortest.....Average.....
15. Average Fired per Charge³.....lb.
16. Depth on Grate at Start of Test.....in.
 (After Firing).....in.
17. Depth on Grate at Finish of Test.....in.
18. Weight as Fired during Test⁴.....in.
19. Weight as Fired per Hour⁴.....lb.
20. Moisture in Fuel.....per cent
21. Weight Fired per Hour less Moisture:⁴
 $\frac{100 - \text{item 20}}{100} \times \text{item 19}$lb.

ASH AND REFUSE

22. Weight of Ash and Refuse Removed from Grate.....lb
23. Weight of Ash and Refuse Removed from Ashpit.....lb.
24. Total Weight of Ash and Refuse Removed⁴
 (item 22 + item 23).....lb.
25. Total Ash and Refuse, Percentage of Fuel as Fired.....
26. Combustible in Ash and Refuse.....per cent

TEMPERATURE

27. Steam.....deg. fahr.
28. Feed Water.....deg. fahr.
29. Gases Leaving Boiler.....deg. fahr.
30. Boiler Room.....deg. fahr.
31. Outside Air.....deg. fahr.

DRAFT INTENSITY

32. In Smokehood.....in. water
33. Over Fire.....in. water
34. In Ashpit.....in. water

OUTPUT

35. Equivalent evaporation from and at 212 deg. fahr. per hr. of test.....lb.
36. Equivalent evaporation from and at 212 deg. fahr. per lb. of dry coal fired.....lb.
37. Heat Recovered at the Outlet per hour (item 35 \times 0.97).....1000 B.t.u.

STEAM AND WATER

38. Steam pressure (gage).....lb. per sq. in.
39. Total Water Fed to Boiler during Test.....lb.
40. Priming: Total Water Removed from Separator,
 Per Cent of Total Feed Water.....per cent

³ When the New fire Method is used the equivalent fuel charged shall be given throughout.
The method of obtaining this is shown at the end of this table.

⁴ See page 11.

⁵ To include ash content of residual fire when *New Fire Method* is used.

HEAT BALANCE

	Per lb. fuel as fired	Per cent heat in fuel fired
41.* Heat to steam leaving outlet (and thermal efficiency boiler, furnace and grate)....
42. Heat lost by hot flue gases, exclusive of steam...
43. Heat lost by not burning carbon monoxide.....
44. Heat lost by steam in flue gas.....
45. Heat lost by combustible in ash and refuse.....
46. Heat lost by radiation.....
47. Undetermined losses and errors.....
48. Total, items 41, 42, 43, 44, 45, 46, 47 and calorific value of dry fuel.....	100

ADDITIONAL ITEMS, FOR USE ONLY WITH NEW FIRE METHOD OF STARTING

FUEL USED

49. Weight of wood for kindling.....	lb.
50. Heat value of wood.....	B.t.u. per lb.
51. Weight of residual fire.....	lb.
52. Heat value of residual fire.....	B.t.u. per lb.
53. Fuel value of wood $\left(\text{item 49} \times \frac{\text{item 50}}{\text{item 11}} \right)$	lb.
54. Total fuel fired during test (exclusive of wood).....	lb.
55. Total equivalent fuel charged during test (item 53 + item 54).....	lb.
56. Fuel value of residual fire $\left(\text{item 51} \times \frac{\text{item 52}}{\text{item 11}} \right)$	lb.
57. Equivalent fuel used during test (item 55—item 56, this value to be used for item 18).....	lb.

ASH AND REFUSE

58. Ash in residual fire (by analysis).....	per cent
59. Total ash content of residual fire: $\left(\text{item 51} \times \frac{\text{item 58}}{100} \right)$	lb.
60. Total ash and refuse removed from ashpit.....	lb.
61. Equivalent ash and refuse removed from ashpit (item 59 + item 60, this is the value to be used for item 23).....	lb.

* Item 41. Heat to "steam," includes the heat used to raise the water removed from the separator from the feed water temperature to the steam temperature.

LOG SHEET NO. 1

GENERAL SHEET

Test of.....boiler with.....coal

Date.....

Test No.....

Time

General Notes

(Here will be recorded the method and times of starting and stopping, the method of firing, the difficulties encountered with ash and clinker, the times of cleaning, slicing and raking the fire, the caking and other properties of the coal, the manipulation of the dampers, etc.)

THE ECONOMICAL UTILIZATION OF HEAT FROM CENTRAL STATION PLANTS

By N. W. CALVERT (MEMBER) AND J. E. SEITER (NON-MEMBER)

DETROIT, MICH.

THE great increases in operating costs during the past decade and the consequent increase in consumers' rates for heat supply have directed the attention of district heating engineers to the need of improved design and more intelligent operation of the consumer's heating system. That there has been room for some improvement in this direction has always been evident, but the great possibilities of economy have only recently been realized. The authors have repeatedly witnessed reductions in heat consumption by improved operation of from 25 to 35 per cent, while in one case a consumer's bills were reduced 72 per cent without discomfort to the occupants of the building. Savings of this magnitude would be possible in many other buildings had certain principles been followed in the original design of the system.

The operation of a steam heating system¹ supplied from a central plant is fundamentally different in several respects from the operation of an isolated plant and system. In the case of an isolated plant, the variation in the amount of heat furnished to a building depends almost entirely on the rate of firing the boiler, whereas with central station heat, there being an unlimited supply, the amount of heat used depends entirely on the method of operation. While this unlimited supply lends a marked flexibility of operation, it also presents a great possibility of waste, inasmuch as with the isolated boiler a positive act, such as putting on fuel or opening drafts, is necessary to maintain the supply of heat, thus increasing the cost of heating; whereas with central heat the positive act is required to cut off the heat supply, thereby reducing the heating cost. This difference may be exemplified by assuming a building being heated and no attention paid to it. If it is served from an isolated plant the fire soon dies and the expense practically ceases, but if it is using central heat, as long as the supply valve is left open, the expense continues.

With an isolated plant the complete utilization of the B.t.u. content of the steam is not essential because the condensate returns to the boiler at a high temperature and less fuel is required to make it into steam again. With central heat, however, complete utilization is essential for economy, as the consumer must pay for every

¹ All of the data and statements in this paper apply to steam heating.
Presented at the Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, New York, N. Y., January, 1924.

B.t.u. which is put into the steam and consequently suffers considerable loss if 10 to 15 per cent of the total heat content of the steam is sent to the sewer with the condensate.

Naturally the greatest saving by proper operation is to be expected during the periods when demand for heat is light, such as mild weather periods, night hours and times when the building is unoccupied. The fixed losses under light load and no load conditions are much less with central heat than with an isolated plant. A slight economy is sometimes effected by varying pressure conditions on the heating system and central service lends itself more readily to pressure variation than an isolated plant.

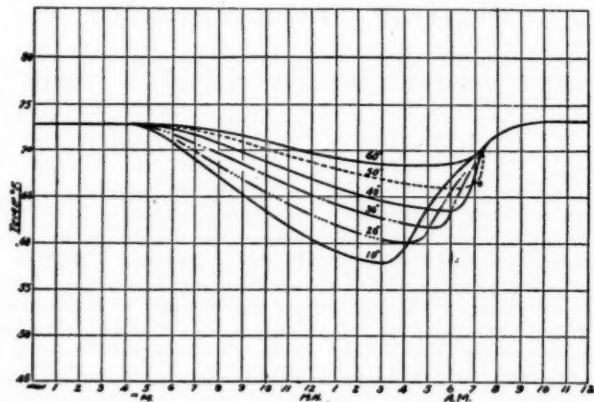


FIG. 1. CURVES SHOWING RATE OF COOLING OF BUILDING INVESTIGATED

The two methods of supply, then, present different problems. In general these problems divide themselves into two classes, *first*, as to proper control of heat supply, and *second*, as to the efficient use of heat. With the above mentioned basic differences in mind, investigations have been conducted by central heating engineers to ascertain the best methods of design and operation of systems supplied with central heat. These investigations, however, have brought to light many facts which apply to either method of supply.

Shutting Off Steam at Night

As has been mentioned, the greatest economy is effected by shutting off steam entirely, especially during the night when the majority of buildings are unoccupied. Inasmuch as the rate of heat transmission from a building is dependent on the differential temperature between the interior and exterior, it is evident that when the interior temperature is reduced heat will not be given up by the building as rapidly as if a high temperature is maintained. In buildings where steam is left on during the night, the interior temperature frequently increases and consequently the loss by transmission through the building walls increases. Investigations have been made of the rate of cooling of various buildings. Fig. 1 shows cooling curves plotted for one building investigated.³ The average building cools very slowly and

³ These curves were plotted from actual tests but have been shifted slightly for the sake of clearness so that they coincide during the middle of the day.

it is possible by a knowledge of the rate of cooling to utilize, during the period in which the building is occupied, a large proportion of the heat put into the building. Referring to Fig. 1, for instance, it is evident that steam might be shut off in this office building an hour or two before closing time without making the building uncomfortable for the occupants, inasmuch as in the coldest weather during the test the rate of temperature drop was only 5 deg. in 3 hr. Fig. 2 shows the saving effected in daily steam consumption by shutting off during night hours.

It should be noted here, however, that a building must have sufficient radiation to heat up readily in the morning, as instances have been found where buildings

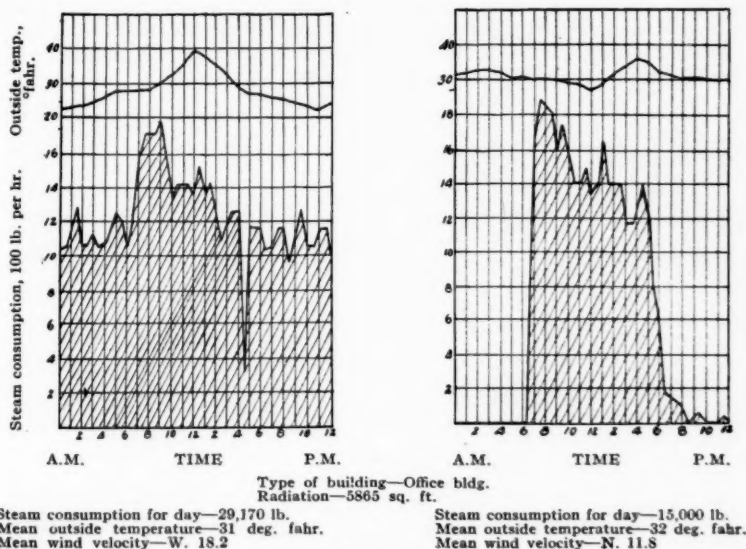


FIG. 2. LOAD CURVE SHOWING SAVINGS POSSIBLE BY SHUTTING OFF AT NIGHT

with insufficient radiation required steam on all night in order to have a suitable temperature during business hours, with consequently increased steam bills. Excessive radiation, on the other hand, also increases the consumption by overheating the building and necessitating the opening of windows.

Shutting Off Steam During the Day

The weather during a heating season might be divided into three classes, mild, cold and extremely cold. The last named generally constitutes only 5 to 10 per cent of the entire heating season, and inasmuch as the heating system is designed to care for extreme conditions, it is logical that some adequate means of control should be provided for use during the periods when these extreme conditions do not exist. In many buildings the heat can be entirely shut off during ordinary weather for many hours of the day as well as at night. Fig. 3 shows savings effected by turning on and off steam at intermittent periods during the day in a large department store without creating an uncomfortable temperature. This

saving is especially interesting when the necessity for adequate heat in such a building is considered.

Installation of a 24-Hour Main

A condition frequently met is that in which, on account of incorrect design of the heating system for central heat, it is necessary to heat the entire building or at any rate a large portion of it for 24 hours a day, because heat is required in one small part of it. This condition is especially true in hotels and clubs, where a lobby must be heated at all times. The same condition exists in office buildings where a small cigar store or an all night restaurant is located. There are also many buildings where the steam for the sprinkler tank is taken directly from the heating system which, therefore, cannot be shut off without endangering the fire equipment. There are garages with a small office to be heated at times when the garage proper needs no heat. Many similar cases could be cited.

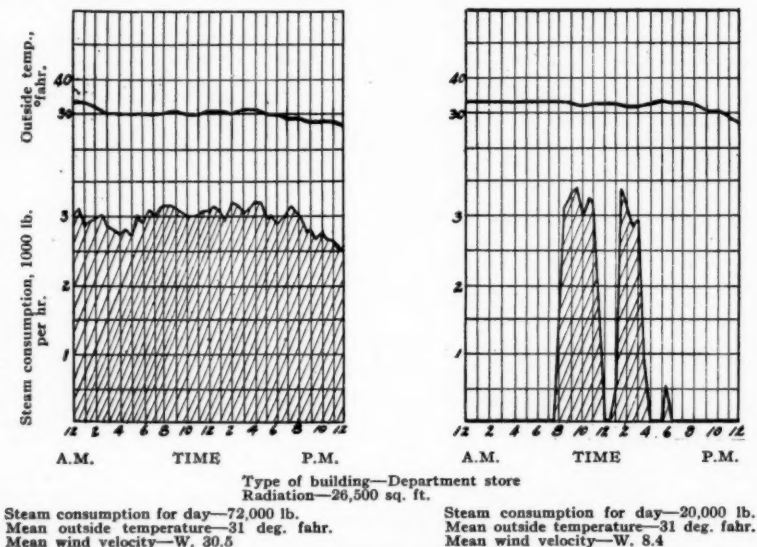


FIG. 3. LOAD CURVES SHOWING SAVINGS POSSIBLE BY INTERMITTENT OPERATION

The remedy for such cases is a separate main run from the service point to the part of the building requiring continuous service. Many Detroit buildings are now being equipped with a 24-hour main which may supply steam to the sprinkler system toilet rooms, in order to protect the plumbing, and any other parts of the building which may have unusual heating requirements. The balance of the heating system may be shut off at all times when the building in general is unoccupied with a resultant economy in operation. Fig. 4 is a diagram of the layout of such a 24-hour main serving in this particular building the sprinkler system, elevator shafts, toilet rooms, lobby and telephone operators' room which are occupied at all times. This main is also designed to take care of the building restaurant requirements and therefore is used during summer months. Needless to say, this arrangement should

be planned when the building is laid out and the rooms requiring continuous service should be kept as close together as possible. Fig. 5 is an example of the saving effected by the use of a 24-hour main, which was installed in an office building to heat the lobby for the night watchman and to heat water for the janitors. This installation, costing about \$200, is making a saving of approximately \$500 per month in cold weather.

Accessibility of Valves

Separate lines and separate valves for operating different parts of a heating system independently according to their demand for heat simplify economical operation,

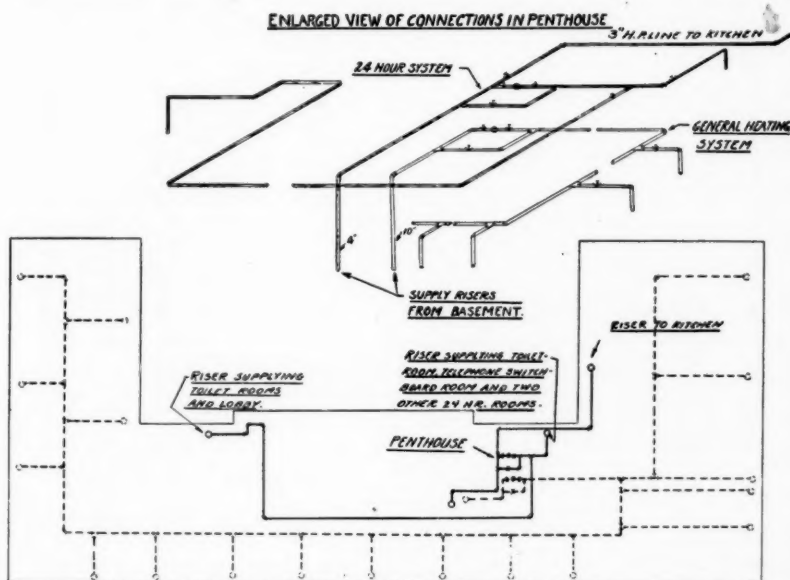


FIG. 4. DIAGRAM OF 24-HOUR SERVICE MAIN

but to make them most successful they must be conveniently located and easy to handle. The operating engineer, no matter how ambitious, will not run from one part of the building to another, closing valves here and there, just to save a little money for someone else. Therefore the installation of valves should be such as to require a minimum amount of time and effort for operation.

Remote Controlled Valves

In many buildings having a number of central heat consumers, each with a separate heating system, the service enters the basement of one store and is distributed from a header to the various other consumers. Naturally it is frequently impossible to give all the consumers access to the basement where the header is located. In such cases, remote controlled valves can be used to advantage. These are of two types, *first*, those consisting of a quick-opening valve with a chain or cable for

operating, and *second*, those which are operated electrically. The first class, using chain or cable, works quite satisfactorily for short and straight distances but the use of pulleys for turns in the line is generally a source of trouble. Good results have been obtained by the use of bell cranks in place of pulleys. The second class, electrically operated, has been worked out successfully by the use of a thermostat motor controlled by a small double-throw switch instead of the thermostat.

Thermostatic Control

In a final analysis, proper temperature control at all times is the key to economical operation of a central heating system. The simplest form of control is that effected by manual operation of valves as has been described. There are also numerous devices on the market for mechanical control of building temperatures. For small

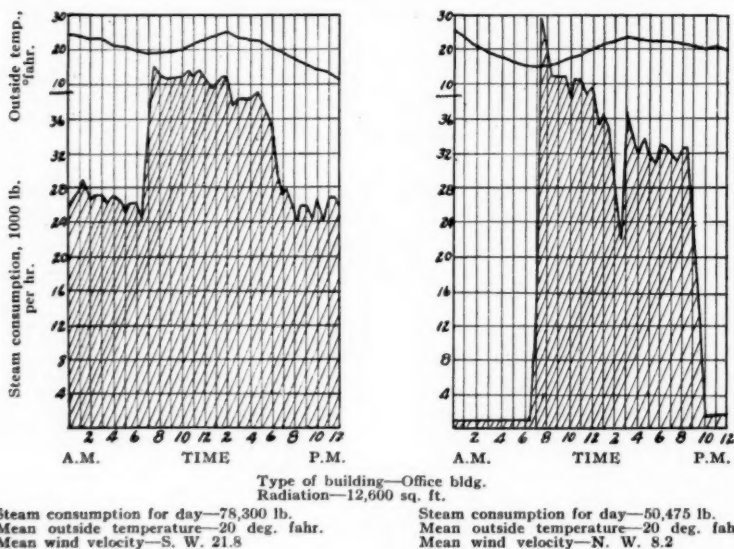


FIG. 5. LOAD CURVES SHOWING SAVINGS POSSIBLE BY MEANS OF 24-HOUR MAIN

buildings the unit thermostat method, in which the steam supply for the entire building is controlled by a single thermostat, is very satisfactory, especially with time attachment for lowering the temperature maintained during periods when the building is unoccupied. Fig. 6 shows characteristic operation and saving effected by unit thermostat control. Some manufacturers of thermostats claim that large buildings are being operated satisfactorily with unit control. While this method may not be impossible for large buildings, it is easy to conceive conditions under which satisfactory control might be difficult.

In the larger buildings the ordinary method of individual control is most desirable in which a number of thermostats are installed in the building, each thermostat operating one or more radiators. However, with individual thermostat control using the "positive"-type thermostat in which the controlled valve is

either wide open or tightly closed, perfect temperature regulation is difficult. When the temperature of the room reaches that set on the thermostat, the valve on the radiator closes. This radiator is still full of steam and the large weight of metal in the radiator itself is at a temperature between 212 and 230 deg. Fahr. and continues to radiate heat to the room. The occupant, becoming too warm, opens the window, thus cooling off the room sufficiently to cause the thermostat to open the radiator valve again.

An intermediate type thermostat, which will open the radiator valve only to a degree required by the need for heat in the room, is much to be preferred, but the present commercial arrangements are not suitable. A very slight lift of an

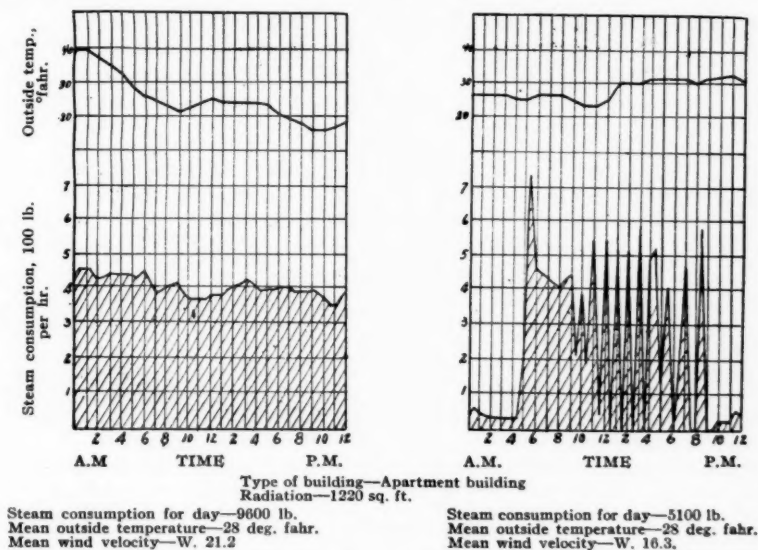


FIG. 6. LOAD CURVES SHOWING SAVINGS POSSIBLE BY "UNIT" THERMOSTATIC OPERATION

ordinary size radiator valve is sufficient to fill a large radiator with steam, and the system then acts the same as with the "positive" thermostat. In one building under the authors' observation in Detroit, orifice sleeves were installed in the radiator valves to restrict the flow of steam to the radiator and these, in conjunction with intermediate type thermostats, give good temperature regulation.

Vacuum Systems

The installation of a vacuum pump on a heating system using central service is a desirable feature from the standpoint of economy of operation. One of the greatest assets of a vacuum system is the positive removal of air which guarantees quick circulation. This insures even distribution of heat throughout the building and permits the building to be brought up to the desired temperature in a minimum time after it has been allowed to cool down. The rapid heating of a building makes it possible to keep steam shut off for longer periods of time. Aside from these

features it has been demonstrated by tests on several buildings in Detroit, that there is a decided saving effected by the use of vacuum pumps because of the increased flexibility of the system. In a straight steam system the temperature of the radiation cannot be varied to any great extent and usually the only means of temperature control is by operating the radiator valves, which is seldom done, and consequently the building is overheated much of the time. By the use of a vacuum system it is possible to vary the temperature of the steam in the radiators considerably, thereby varying the rate of heat transfer. The possible variation may be at least 30 per cent from 230 deg. fahr., corresponding to a steam pressure of 6 lb. to 170 deg. fahr., corresponding to a vacuum of 17 in.

To secure economy by the use of a vacuum system, proper maintenance of the system is absolutely essential. The system should never be allowed to get in such

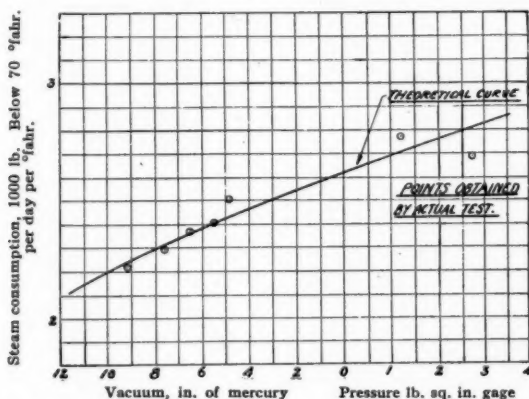


FIG. 7. CHART SHOWING VARIATIONS IN STEAM CONSUMPTION FOR VARIOUS PRESSURES AND VACUUMS AT REDUCING VALVE

a condition as to make jet water necessary to condense steam and vapor in the return lines. Jet water is very undesirable and is unnecessary in a properly maintained system.

If the building engineer avails himself of the opportunity afforded by a vacuum system to regulate the temperature of the steam by varying the pressure, a considerable saving can be effected. The distribution mains of a heating system are usually in the attic or basement spaces which require little or no heat. Even though the piping is covered with ordinary covering there still is considerable heat loss from these mains. Reducing the temperature of the steam by varying the pressure reduces this loss. With a vacuum system considerably greater pressure drops through the system are possible with a low initial pressure and the size of the piping may be reduced, thus cutting down the cost of the installation and the radiating surface of the piping.

A vacuum system has some disadvantages which may be cited. The initial cost is greater than some other systems; it needs more attention than open systems; the maintenance cost is higher; and the desirability of duplicate pumps requires more floor space. Results of investigations seem to show, however, that these objections are of minor importance.

Complete tests to determine the economy effected by vacuum systems were conducted on two buildings, one having 6400 sq. ft. equivalent direct radiation and the other 40,000 sq. ft. The tests consisted of operating under pressure and vacuum conditions for alternate weeks throughout the heating season and observing results. The heating systems in both buildings are vacuum return and are equipped with thermostatic traps on the radiators and riser drips. When operated under a vacuum the reducing valve on the steam supply was set so as to maintain a pressure below atmosphere in the radiators and piping. In the smaller building, the vacuum pump was installed recently and a comparison of the steam consumption before and after is possible.

Total consumption, Oct. 1, 1921 to May 1, 1922 (before installation).....	3,642,505 lb.
Total consumption, Oct. 1, 1922 to May 1, 1923 (after installation) corrected to temperatures for 1921-22....	3,297,500 lb.
Saving in 7 months operation with vacuum.....	345,000 lb.
Average saving per month.....	50,000 lb.
Estimated total saving for season, 8 months.....	395,000 lb.
Estimated saving in dollars, with steam at \$1.00 per 1000 lb.....	\$395.00
Total actual cost of pump and piping installed.....	\$658.00

These results are self explanatory, and the saving is very evident.

The larger building has individual thermostat control. The equipment in the large general office rooms is of the positive type in which the radiator valve is either wide open or closed. In the private offices the thermostats operate modulating valves permitting the radiators to heat fractionally. The results of the tests in the larger building are shown in Fig. 7. The points shown are averages for periods of about five days each. Some difficulty was encountered due to the wide variation in outside temperature in holding a constant pressure over extended periods. If the weather became very mild during a period scheduled for operation with pressure above atmosphere, or if it became exceptionally cold during a period scheduled for operation with vacuum, it was practically impossible to maintain proper building temperature without varying the pressure. For this reason there are no points available for the curve between 1 lb. pressure and 4 in. vacuum as on the days when the system was being operated at these points the test was interrupted.

One objection advanced to vacuum systems was that due to the pressure differential at the thermostatic trap, vapor would be allowed to pass into the return lines. Recording gages were installed to check the differential pressure and leakage was carefully watched for, but no appreciable amount of vapor was noted at high vacuums. No cold water jets were used nor were necessary to maintain a high vacuum. No vapor was ever noticed at the meter. The drop across the trap can be as low as desired. In Fig. 8 is shown diagrammatically the pressure drop through a vacuum system. The upper curve shows the condition with 5 lb. pressure at the reducing valve and 12 in. vacuum at the pump. The middle curve has atmospheric pressure at the reducing valve and 12 in. vacuum at the pump. The lower curve, having a 10 in. vacuum at the reducing valve and 12 in. vacuum at the pump, is typical of the conditions under which the tests which have been described were run, and statements made as to savings possible apply only to these or similar conditions where a pressure below atmospheric is maintained at the reducing valve.

Tests on a number of other buildings showed an average saving of about 28 per cent effected by variation of steam temperature by means of a vacuum pump.

Incidentally the marked savings obtained by variation of radiator temperature by control from one central point directs attention to the merits of the hot-water heating system in which a wide range of radiator temperatures is possible. It is quite possible that the better regulation and consequent economy of the hot-water system may overbalance the greater first cost as compared with a steam system.

Building Details for Efficient Heating

In the construction of the building itself many features may be incorporated which may simplify economical operation. Insulating materials may be used in

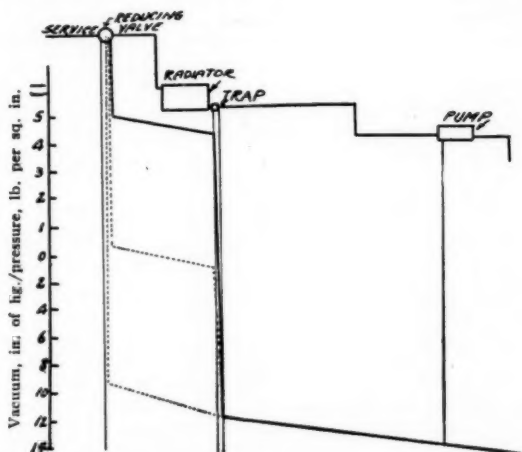


FIG. 8. PRESSURE DISTRIBUTION OF VACUUM HEATING SYSTEM WITH VARIOUS PRESSURES AT THE REDUCING VALVE

the walls and roof. Window and door frames should be made as tight as possible and weather strips installed, if necessary. Revolving doors to counteract the stack effect of large buildings are a worth while investment. Automatic doors on garages and similar buildings have shown decided savings by actual test. The advantages of these details of construction are generally known but they are seldom installed purely for benefit to the heating system. This is a deplorable fact as the heating requirements should be taken into more serious consideration in the design of a building. However, the more immediate problem of the heating engineer is the design of the heating system itself.

Salvaging Heat in Condensate. The heat of the liquid constitutes about 15 per cent of the total heat of the steam used for heating buildings and there is little of it given up by the condensation in most types of heating systems. In many central heating systems where the condensate is not returned to the boiler plant but is permitted to go to the sewer before this heat is extracted, a considerable waste occurs. To utilize a portion of this otherwise wasted heat the installation of some sort of economizer is very desirable. These economizers are divided into several

classes according to their construction and use as indirect air heating, direct air heating and water heating economizers.

Indirect Air Heating Economizer. The indirect air heating economizer is usually installed as shown in Fig. 9. It consists of a coil enclosed in a sheet metal box in the basement with a register in floor leading to the room above. The air inlet is arranged to take air from the basement or from outside. While this type of economizer is quite efficient, the cost of installation is rather high in proportion to the economy effected.

Direct Air Heating Economizer. During the past two years experiments have been conducted on a direct air heating economizer installed as shown in Fig. 10. The application of this particular economizer, however, is limited to heating systems having sufficient pressure in the return lines to raise the condensate to the economizer. A similar arrangement can be used in places where a basement room is to be heated and, as it necessitates no lift, can be installed on any type of system.

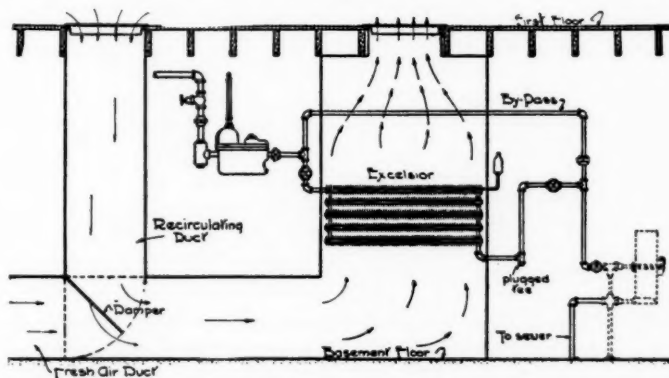


FIG. 9. DIAGRAM OF INDIRECT ECONOMIZER INSTALLATION

An economizer as shown in Fig. 10 having 60 sq. ft. of surface was installed in a building having 545 sq. ft. of radiation. Test data showed that condensate entered the radiator at 175 deg. fahr. and discharging at 110 deg. fahr. and that 9300 B.t.u. were salvaged per average day (outside temperature 38 deg.). This would represent an annual saving of 25,000 lb. of steam, or at \$1.00 per thousand, \$25. The cost of the installation approximates \$75.

Water Heating Economizer. The majority of buildings use hot water in sufficient quantities to warrant the installation of a storage tank containing adequate heating surface so that by passing the condensate through this tank, the heat may be transferred to the water. Such an installation constitutes the water-heating economizer.

In the selection of an economizer, obviously the variation in the heat available in the condensate, the water requirements, the amount of heating surface, the amount of storage capacity, the actual return on the investment, and the proper piping layout must be considered.

The recommended method of installation of a water heating economizer is shown in Fig. 11. A live steam booster is placed above the economizer so as to heat the

water to the temperature required throughout the building. This temperature is regulated by a thermostatic valve on the steam main. The returns from this heater also pass through the economizer. The cold water enters the bottom of the economizer and is heated by the condensate. This tempered water leaves the top of the economizer and passes through the booster into the system. If recirculating lines are installed on the hot-water system they should be connected to the bottom of the booster and not the economizer, otherwise the storage in the economizer will be filled with hot water heated by the booster and the effectiveness of the economizer impaired.

Detailed tests were made on such an economizer. In Fig. 12 is plotted performance of the economizer against daily steam consumption in the building. By performance is meant the ratio of the heat transfer in the economizer to total heat in the condensate above 32 deg. fahr. This curve shows that with a given amount

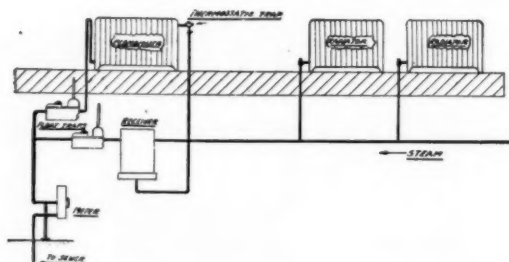


FIG. 10. LAYOUT OF DIRECT AIR HEATING ECONOMIZER

of heating surface the effectiveness of the economizer in removing the heat from the condensate continually decreases as the steam consumption increases.

Fig. 13 shows the value of the heat salvaged had this particular economizer been installed in buildings using various amounts of steam. It can readily be seen that the value of heat salvaged increases very rapidly as the steam consumption increases until the limit of the economizer is reached. After this limit is reached, the value of heat salvaged increases much more slowly in proportion to increase in heating requirements.

The characteristic curves of this economizer showed very clearly that there is some particular size of economizer which will show the highest return upon the investment. From the data obtained it was possible to compute the most economical amount of heating surface to be installed in economizers used on systems having various heating requirements. The results of these computations are shown by the upper curve on Fig. 14.

From a study of the water temperature entering and leaving the economizer, it was seen that during the periods of little water flow, heat was given up to and stored in the tank. This study also showed that the storage capacity of the economizer tested could be increased because the cooling of the condensate gradually diminished during a period of high condensation flow and low water flow. Just what the storage capacity should be is difficult to estimate, but with the ratio of 1 to 3 of storage capacity to hot water consumption per day, as in this particular economizer, there is a great saving on the investment and this is a safe ratio.

In Fig. 14 are plotted, in addition to the heating surface, the value of heat salvaged, annual cost of hot water equipment chargeable to the economizer and savings effected. The savings possible are governed by the relation of heating surface to condensate flow and storage capacity. The storage tank, piping and labor are the expensive parts of any economizer installation, and additional heating surface does not increase the total cost of the installation materially. For ready reference Table 1 has been prepared from Fig. 14. The savings shown are over and above the annual cost of the economizer which is figured as 20 per cent of the initial cost—a very liberal assumption.

These investigations show that a water heating economizer is a paying investment on any size heating system if there is sufficient hot water demand to utilize some of the available heat in the condensate, and if the temperature of the returns is not too low due to the use of atmospheric or vapor systems. However, care must be

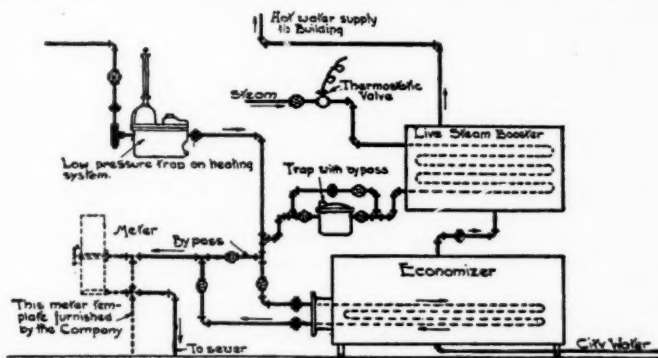


FIG. 11. METHOD OF INSTALLING WATER HEATING ECONOMIZER

taken to provide ample heating surface and storage capacity in order to have an efficient system.

Atmospheric and Vapor Systems

In connection with the subject of salvaging heat from the condensate, atmospheric and vapor systems and installation of radiator traps are to be commended, as they eliminate steam from return lines and discharge the condensate to the sewer at a low temperature. The installation of modulating valves such as are generally used on atmospheric and vapor systems, however, does not seem to be of any particular advantage. While the valves are all right theoretically, an inspection of several hundred such valves on an average winter day disclosed 45 per cent wide open, 50 per cent closed and only 5 per cent partially open. In other words, in only about 5 per cent of the cases did the occupants actually take advantage of the regulating feature of the valve. The buildings investigated were stores and office buildings.

Data on Heat Consumption

In the course of the investigations conducted in connection with economical utilization of heat much valuable data of a general nature has been compiled. Fig. 15 shows the variation by months of steam consumption in Detroit by all

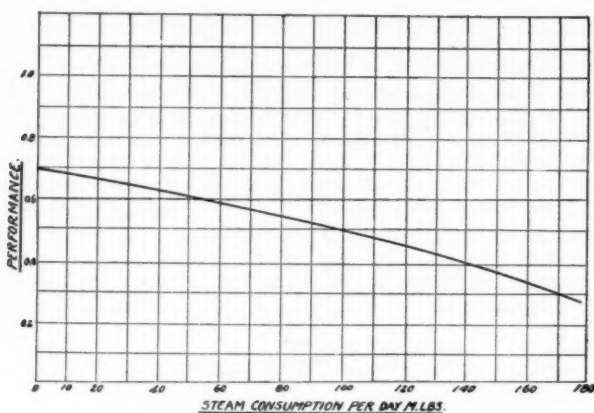


FIG. 12. CHART SHOWING PERFORMANCE OF ECONOMIZER

Note: Performance—Ratio of heat transferred to total heat in condensate above 32 deg. Fahr.

classes of customers for various outside temperatures. Note that this is a straight line, showing that the consumption varies directly with the difference between inside and outside temperatures. It is an interesting point that the 1922 figures all fall below the line, as it was in 1921-22 when an intensive educational campaign was undertaken.

As the curve in Fig. 15 is a general average for all types and sizes of buildings, factors were computed from a study of 173 Detroit buildings, each having over 4000 sq. ft. of radiation. These factors take into consideration the required in-

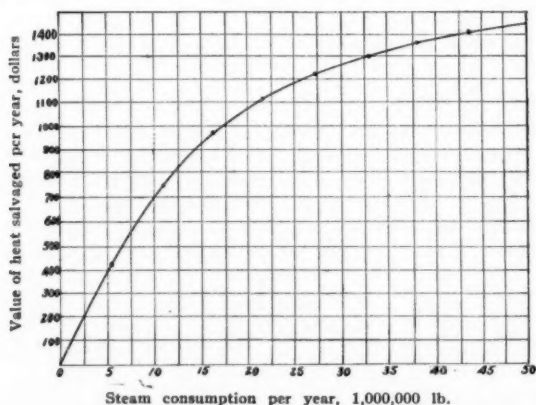


FIG. 13. CHART SHOWING VALUE OF SALVAGED HEAT FOR SAME SIZE ECONOMIZER INSTALLED IN BUILDINGS WITH VARIOUS STEAM CONSUMPTIONS

Note: Value of heat salvaged computed with steam at \$1.00 per 1000 lb.

side temperatures and hours of use for various types of buildings. To estimate the probable steam consumption in any particular building the reading on Fig. 15 should be multiplied by the factor for buildings of its type as given in Table 2.

The proportions of total annual consumption in Detroit for each month is as follows: October—7 per cent, November—12 per cent; December—16 per cent;

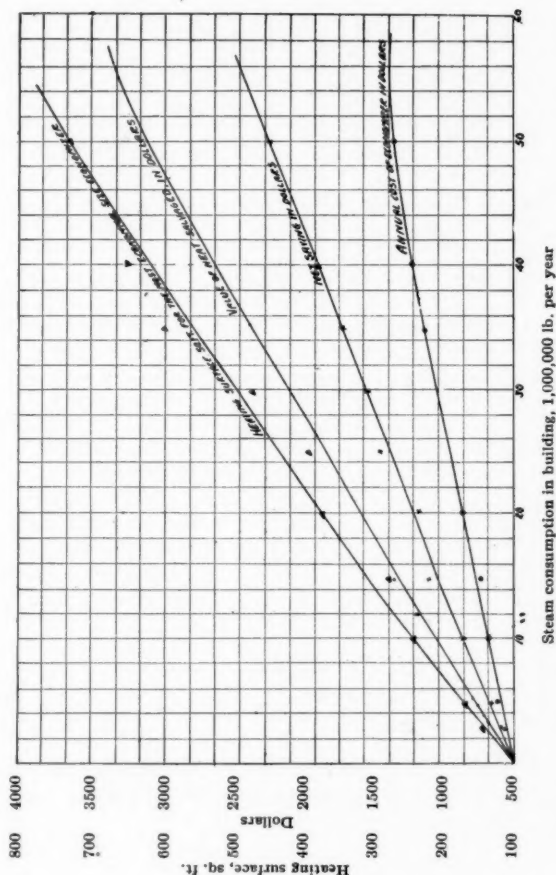


FIG. 14. CHART SHOWING ANNUAL COSTS OF ECONOMIZERS, GIVING VALUES OF HEAT SALVAGED AND SAVINGS WITH ECONOMIZERS OF THE SIZES SHOWN FOR BUILDINGS WITH VARIOUS STEAM CONSUMPTIONS

Note: Cost of steam estimated at \$1.00 per 1000 lb.

January—18 per cent; February—18 per cent; March—15 per cent; April—9 per cent; May—5 per cent. It must be borne in mind, however, that the figures shown apply to conditions in Detroit and will vary for other cities.

Examples of savings made in Detroit buildings of various types are shown in Table 3. The actual consumption with economical operation is corrected to temperature of period of average operation.

TABLE 1. ECONOMIES POSSIBLE IN SALVAGING HEAT BY USE OF ECONOMIZERS

Steam consumption per year, lb.	Radiation sq. ft. (est. 600 lb., yr. sq. ft.)	Heating surface in econ., sq. ft.	Annual cost of econ., dollars (est. 20% total cost)	Value of heat salvaged, dollars, steam cost \$1.00 per 1000 lb.	Saving per year, dollars net
100,000	170	4	7.00	8.00	1.00
200,000	330	6	14.00	15.00	1.00
300,000	500	8	21.00	22.00	1.00
500,000	830	12	34.00	37.00	3.00
750,000	1250	14	37.00	54.00	17.00
1,000,000	1670	15	39.00	69.00	30.00
1,500,000	2500	25	50.00	115.00	65.00
2,000,000	3330	36	65.00	150.00	85.00
3,000,000	5000	60	98.00	228.00	130.00
5,000,000	8330	100	164.00	382.00	218.00
7,000,000	13330	150	200.00	570.00	370.00
10,000,000	16670	200	257.00	762.00	505.00
15,000,000	25000	250	320.00	1168.00	848.00
20,000,000	33330	385	510.00	1480.00	970.00
25,000,000	41670	410	530.00	1870.00	1340.00
30,000,000	50000	525	680.00	2170.00	1490.00
35,000,000	58330	700	925.00	2650.00	1725.00
40,000,000	66670	775	1020.00	3020.00	2000.00
45,000,000	75000	812	1115.00	3360.00	2245.00
50,000,000	83330	900	1200.00	3680.00	2480.00

TABLE 2. FACTORS TO BE APPLIED TO STEAM CONSUMPTION CURVE FOR VARIOUS TYPES OF BUILDINGS

Hotels.....	Average of 13 buildings	1.35
Public Buildings.....	Average of 4 buildings	1.24
Garages.....	Average of 4 buildings	1.24
Wholesale Houses.....	Average of 3 buildings	1.24
Printing Houses.....	Average of 6 buildings	1.22
Office Buildings.....	Average of 34 buildings	1.04
School Buildings.....	Average of 4 buildings	.94
Auto Sales and Service Bldgs.	Average of 4 buildings	.91
Loft Buildings.....	Average of 6 buildings	.87
Clubs.....	Average of 5 buildings	.79
Stores.....	Average of 29 buildings	.79
Warehouses.....	Average of 4 buildings	.75
Banks.....	Average of 9 buildings	.67
Theatres.....	Average of 13 buildings	.48
Churches.....	Average of 6 buildings	.44
Dance Halls.....	Average of 3 buildings	.39

In the nineteen cases shown in Table 3, the average saving was 31.3 per cent. As these examples were selected almost at random the possibilities for economy in the average building are quite evident.

Summary

In summing up the subject of economical utilization of central station heat, the following fundamentals are tabulated:

1. Installation of sufficient and properly placed radiation
2. Shutting off for maximum possible time at night and during day
3. Salvaging the heat in the condensate
4. Proper temperature control
5. Reduction of transmission loss.

If the factors mentioned are given proper consideration in the design and operation of the heating system, maximum comfort at a minimum cost is practically assured.

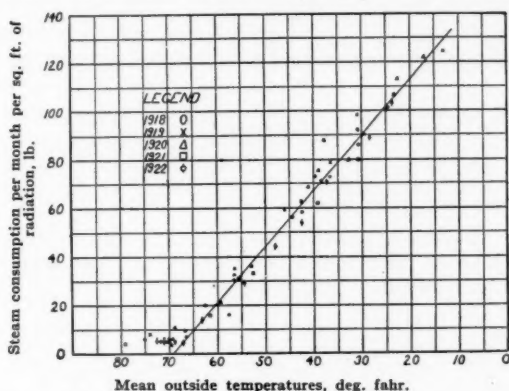


FIG. 15. SALES OF STEAM PER SQUARE FOOT OF RADIATION PER MONTH FOR VARIOUS OUTSIDE TEMPERATURES, CORRECTED TO A 30-DAY MONTH

TABLE 3. EXAMPLES OF SAVINGS EFFECTED BY ECONOMICAL OPERATION

Building	Average Consumption per Day, Lb.			
	Radiation	Before investigation	After investigation	Saving %
Office Bldg. No. 1	17600	53746	42546	20.9
Office Bldg. No. 2	15300	44309	39300	11.3
Office Bldg. No. 3	10200	56400	31577	44.0
Store Bldg. No. 1	1990	4986	3874	22.3
Store Bldg. No. 2	77600	185150	82625	55.4
Store Bldg. No. 3	8740	21325	9660	54.7
Store Bldg. No. 4	2580	7267	4340	40.3
Store Bldg. No. 5	5600	16967	11467	32.4
Church No. 1	1150	4963	3150	36.5
Church No. 2	11410	8010	5930	25.9
Theatre	13750	13176	11180	15.1
Club Bldg.	2893	6142	3992	35.0
Auto Sales and Service Bldg.	1175	6618	3683	44.4
Apartment Bldg.	1834	8767	7171	18.2
Garage	14700	49767	39477	20.7
Public Bldg.	53500	68529	54531	20.4
Bank Bldg.	21555	45323	29918	33.8
School	9160	29705	21955	26.0
Warehouse	5920	21130	13330	36.8

DISCUSSION

M. W. EHRLICH: I am particularly interested in the statement on vacuum heating, showing savings around 28 per cent. However, there are some points on which one should have more information to interpret the data presented. For instance, there is no initial pressure noted on the supply end. If it is assumed

that most central stations operate above atmosphere on the supply mains, then it would be interesting to know how they obtain such high minus pressure or vacuum at the reducing valve, from a high-pressure supply, and keep it so close as indicated.

The question of proper radiation was emphasized, and an interesting point to be considered is what transmission values are considered correct for the right amount of radiation.

Thermostatic traps leaking steam, in some cases due to pressure drop, was a point mentioned. Such leakage would mostly be due to re-evaporating when discharging from the higher pressure vacuum condition. With a proper relation of pressure on the inlet and vacuum on the other end, also a good working trap, noticeable leakage would not occur.

It would be interesting to learn how the 17 in. of vacuum are maintained on the return end, and also if the returns go back in a return main going to the power house, or if they are discharged into the sewer.

G. B. NICHOLS: It is surprising to learn that within one year or possibly two what an outgrowth of central heating plants there has been. In New York progress in this particular line has been rather slow. As an illustration, there is one central heating plant company in New York City, operating two plants. The great office center at the present time is not tapped in any way by a central heating plant, which is the greatest opportunity of its kind in the country. The Middle West is building plants all over and is far ahead in this work.

In no time during the building periods of this country have opportunities of this kind seemed so advantageous. For instance, at Jackson Heights a suburb of New York City where there are approximately 2000 apartment houses built under one management, all with individual plants, a greater opportunity could not have been offered for building central heating plants. Realization of the advantages of such a plant is just beginning to dawn, and there is one under construction at 181st St. where 15 apartment houses will all be heated by a central hot water system with central refrigeration. There is no question that the apartment tenants are wasteful of heating service, and anything that can encourage that is bound to produce these central heating plants.

J. R. McCOLL: I would like to ask Mr. Calvert or Mr. Seiter if they would recommend a design of the radiation on one side of the building, for instance, the windward or northern side, to be separately controlled from the radiation on some other side or in courts, so that the radiation in these various parts of the buildings can be shut off at different periods. That is a matter of particular interest to designers. Then I would like to ask what would be their recommendations as to periods of operation in a plant where the heating is done by hot water heated by central steam. Could the temperature of the water be controlled, and therefore the temperature of the building, say for 24 consecutive hours, or would intermittent service be better?

This is one of the most valuable papers presented before the Society in many years and the authors are to be congratulated on their fine work.

H. M. HART: I could not understand how the condensation from this receiver went through the economizer when they had a pressure on, and when they didn't have a pressure on it automatically went out through the float trap. Even when the pressure was on, I could not understand why it would not go through the float

trap. Perhaps there are some valves that ought to be put in there, and are not shown in the illustration, Fig. 10.

W. T. JONES: One of the objections in shutting off steam is that it tends to develop leaks in the system. Would the authors explain whether leakage would have any effect and to what extent the continuous shutting off the heat would tend to cause leaks and expensive repairs?

H. M. HART: In trying to maintain high vacuum condition throughout a system of some size it has been found that very often there is not sufficient volume of steam to fill the entire system and the tail ends of the system would be without steam when the nearer portions to the source of supply would be comfortably heated.

HELEN R. INNES: Is there much variation in building temperatures during hours of occupancy with the intermittent operation? When some pressures are dropped during hours of occupancy, is there noticeable effects in the temperatures of the rooms?

N. W. CALVERT: The question was asked, how we get a vacuum down to 17 in. and probably about 10 in. at the pressure-reducing valve. With the pumps that we use very good vacuums can be obtained and in some cases difficulty is met with in trying to control the pressure-reducing valves closely enough, and in that case, throttle valves have been installed, which are used in conjunction with the pressure-reducing valves. Incidentally, it might be stated that the distribution pressure on our lines is in the neighborhood of 25 to 30 lb. Regarding the question of the drop in pressure through the thermostatic traps, the experiments presented have involved a vacuum throughout the entire system with a very small differential if any, across the radiator trap. Concerning the discharge of the returns, in most of the heating systems the returns are being discharged directly to the sewer, and that is what makes the question of economy a very vital factor.

M. W. EHRLICH: We should know the transmission valves referred to that would be considered as giving the proper amount of radiation; in other words, as the whole subject of economy depends largely upon the proper amount of radiation, there must be some transmission factor that would pretty nearly indicate the right amount.

N. W. CALVERT: The exact figures are not at hand, but in the slides shown, having the theoretical curve worked out, the points on that theoretical curve were taken from the book by Webster on Steam Heat, giving the varying rates of transmission for the different temperature differentials. The particular point to be shown here is that the actual test points fall very close to the theoretical line as given in the table. Another question was whether the design of heating systems should be modified for intermittent operation, such as separately controlling different parts of the building. The answer is decidedly affirmative. For instance, a nineteen story office building has an overhead supply for the greater part of the building and has a basement supply for the first and perhaps second floor, and the risers are all valved separately. The engineer on the job has some long-distance indicating thermometers which he can read at his desk in the engine room and find out the temperature in any part of the building. If he learns that due to a cold wind one side of the building is cold, or because of the effect of the sun another side of the building is unusually warm, he has some of the risers on the affected side shut off and by that means he accomplishes a very great economy.

In reply to the question of periods of operation for hot-water, our work has covered only work on hot-water systems heated from a central station. It would seem that the economical way of operating such a system would be the same, as is used in operating a district hot-water system, and that is varying the temperature of the water in some way, to conform with the outdoor temperature.

In connection with vacuum heating this vacuum need not be pulled on a building at all times but the engineer should work out a scale whereby he can vary his conditions of pressure and vacuum to take care of outdoor conditions. It is not necessary to design a system that will operate with 17 in. of vacuum and give adequate heat in zero weather, but with a pressure system that is designed with vacuum returns, it is possible to pull a vacuum on that system in mild weather, thereby reducing the temperature of the radiation and effecting considerable economy.

Answering the question about this peculiar economizer, no valves are needed for operation other than those shown in the illustration, but for further elucidation, when the pressure is insufficient to raise the water up to the economizer, the separator or receiver fills to a level with the lower part of the pipe and it overflows.

Concerning leakage caused by the shutting off and turning on of steam, that problem has never come to our attention before. So far as is known, there has never been any amount of leakage from that cause. The question of the volume of steam being reduced to a very small amount at the end of the lines resolves itself into a question of proper pipe sizes.

About the variation in temperatures in hotel buildings with intermittent operation, if the buildings are apartment hotels and bachelor quarters unoccupied during the day there is no necessity of keeping them at a high temperature. The method of operation used is that steam is turned on about 5 o'clock in the morning, and after the building is vacated about 8:30 or 9 o'clock, the steam is shut off and left off as long as possible without endangering the plumbing or causing too much discomfort to the people who are working in the building. If the inside temperature gets too low, steam is turned on for an hour or so and then shut off again. About 4 o'clock in the afternoon the heat is turned on again and left thus until 10 or 11 at night, at which time it is shut off, and the building will probably remain comfortable until the majority of occupants have retired.

AN IMPROVED METHOD OF DETERMINING THE HEAT TRANSFER THROUGH WALL, FLOOR, AND ROOF SECTIONS

By R. F. NORRIS¹, H. H. GERMOND² AND C. M. TUTTLE³, MADISON, WIS.

NON-MEMBERS

IN determining the thermal conductivity of a material, the general method has been to measure the amount of heat supplied to a body per unit time. This body is surrounded by the insulator under test which in turn is surrounded by a cold body. The temperature drop across the insulator is measured. Then if the flow of heat is uniform throughout the insulator, *i. e.*, if equal amounts of heat flow across the insulator per unit area, the thermal conductivity K equals the total heat input per unit time divided by the product of the temperature drop across the insulator and the total area of the insulator under test.

$$K = \frac{Q d}{(T_h - T_c) A t} = \text{conductivity in B.t.u. per hour per deg. fahr. per sq. ft. for 1 in. thickness.}$$

Where Q = heat input in B.t.u.

T_h = temperature in fahr. degrees of hot surface

T_c = temperature in fahr. degrees of cold surface

A = area of insulator in square feet

d = thickness of material in inches

t = time in hours

The conditions prescribed are most nearly fulfilled when the insulator is in the form of a spherical shell of large diameter in proportion to the thickness of the shell. Obviously, this form can be used only for certain types of loose materials; it has been successfully employed by Nusselt in determining the thermal conductivities of asbestos, kiesulguhr, etc.

In the case where the heat flow is not uniform across all portions of the insulator, but where the distribution on the heat flow may be accurately determined (by experiment or calculated) the conductivity may be likewise determined by making the same measurements as before.

The Bureau of Standards has outlined a method of testing insulators of medium

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thickness. Two samples of the same size and thickness are placed on either side of a flat heating unit and the whole is placed between two waterbacks. The heating unit is divided into an inner square and an outer guard ring. The current through the guard ring is enough larger than that in the central square to compensate for edge losses. The electrical input per unit area to the inner square is measured and the temperature difference is determined by means of thermocouples (used in conjunction with a potentiometer). (TRANSACTIONS, Vol. 26, 1920, p. 566.)

There are many difficulties in the application of any of these methods to determine the thermal conductivities of walls. The possibility of using any wall section other than flat is obviously out of the question, thus precluding the use of many of the methods so far outlined. The method used by the Bureau of Standards on insulating material is not readily applicable to the testing of walls, since for a wall section small enough to permit of easy handling, the edge loss is far from inappreciable. Furthermore, it is desirable to have only one wall under test instead of two.

Method Developed in Burgess Laboratories

When there is uniform heat flow through a material in a direction perpendicular to its surfaces, the conductivity is given by the equation:

$$K = \frac{Qd}{(T_1 - T_2) At}$$

Where Q = heat input in B.t.u.; d = thickness; T_1 = temperature hot face; T_2 = temperature cold face; A = area; t = time in hours.

A material which has a constant conductivity may be used as a "heat meter" or thermal "wattmeter" once its conductivity has been accurately determined. Thus, if a piece of that material of thickness d and of fairly large area be placed between a hot reservoir and a cold reservoir, then the rate of heat flow (after equilibrium has obtained) is:

$$\frac{Q}{At} = \frac{K (T_1 - T_2)}{d} = \text{B.t.u. per hour per sq. ft.}$$

If this material of known conductivity K_s be placed face to face with a material of unknown conductivity K_x and placed between two surfaces which are maintained at different constant temperatures, K_x may be determined by observation of three temperatures (or two temperature differences) only.

Let T_1 = temperature at surface against known material
 T_2 = temperature at surface between known and unknown material
 T_3 = temperature at surface against unknown material.

$$\text{Then } \frac{Q_s}{At} = K_s (T_1 - T_2) = \frac{Q_x}{At}$$

$$\begin{aligned} K_x &= \frac{Q_x d_x}{(T_2 - T_3) At} = \frac{Q_s}{At} \frac{d_x}{(T_2 - T_3)} \\ &= \frac{(T_1 - T_2) d_x}{(T_2 - T_3) d_s} K_s \end{aligned}$$

Therefore

$$K_x = \frac{(T_1 - T_2)}{(T_2 - T_3)} \frac{d_x}{d_s} K_s$$

In the case of walls, the conductivity is stated for the thickness of the wall rather than for a unit thickness, and the above equation becomes:

$$K_w = \frac{T_1 - T_2}{T_2 - T_3} \frac{K_s}{d_s}$$

This method may be likened to placing a known and an unknown resistance in series in an electrical circuit and determining the unknown resistance by comparing the voltage drops across both.

This method obviates the need of any measurement of the input to the heating unit. The next step is either to compensate for, or else eliminate the edge loss.

Experiments have shown that it is possible to calculate the edge loss fairly closely for known conditions of edge insulation, average temperature difference between wall and air, etc. However, as these edge losses may amount at times to over a third of the total heat entering one face of the wall, such method might be open to considerable criticism.

By placing a standard thermal resistance (*i. e.*, a flat material of predetermined conductivity) on each side, the heat flowing into the wall and the heat out on the other side are both measured and the edge loss is directly determined as the difference between the input and output.

The edge loss can be reduced considerably by packing suitable insulating material around the edges, but the edge loss will not be reduced to zero even when a considerable bulk of edge insulation is used. Since the problem is to prevent the wall section losing heat along its edges, and since it appears impracticable entirely to prevent heat from flowing through the insulation, the logical answer appears to be to provide something other than the wall to supply this heat. This is accomplished by setting the wall section in a guard ring—an electrically heated frame extending around the four edges of the wall section. After a little practice it is possible so to adjust the input to the guard ring that the heat flow intensities through the two standard thermal resistances are practically the same.

$$\text{Let } q_1 = \frac{Q}{At} = \frac{K_{s1}}{d_{s1}} (T_1 - T_2)$$

$$q_2 = \frac{Q_2}{At} = \frac{K_{s2}}{d_{s2}} (T_3 - T_4)$$

$$\text{Then } K_{wall} = \frac{q_1 + q_2}{2} \frac{1}{(T_2 - T_3)}$$

$$\text{When } \frac{K_{s1}}{d_{s1}} = \frac{K_{s2}}{d_{s2}} \text{ let them both equal } K_s$$

$$K_{wall} = \frac{(T_1 - T_2) + (T_3 - T_4)}{2 (T_2 - T_3)} K_s$$

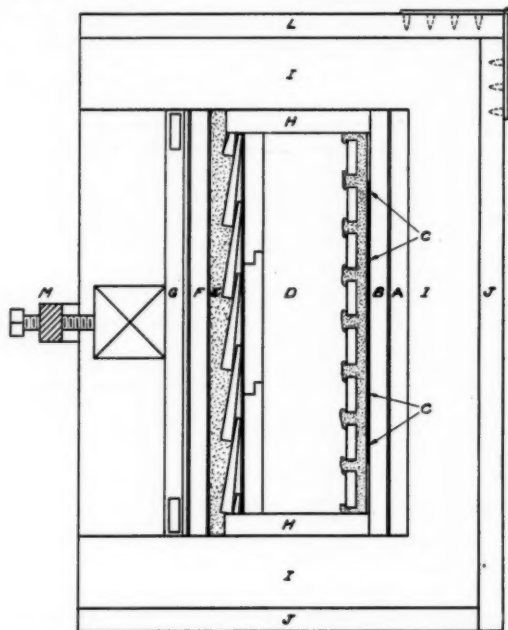
The arithmetic mean of $(T_1 - T_2)$ and $(T_3 - T_4)$ is taken to accommodate any small difference which may exist between two valves; $\frac{q_1 + q_2}{2}$ is thus taken as the mean heat-flow intensity through the wall.

Description of Apparatus Used. The illustration shows (Fig. 1) a cross section through the apparatus used.

The apparatus is housed in a box about 18 in. deep, 36 in. long, 26 in. high,

the top and one end of the box being hinged to permit of easier access. The box is lined with a number of layers of Balsam wool to reduce the heat loss as much as is practicable.

A large flat electrical heating unit *A* is placed at the back of the box. In front of this and close against it, is placed the first heat meter *B* (described elsewhere). A flat thermometer *C* composed of fine insulated copper wire fastened to one surface of a piece of micanite (12 x 18 in.) is placed concentric with the heat meter,



- | | |
|---|--|
| <i>A</i> Main heating unit | <i>G</i> Waterback |
| <i>B</i> "Heat Meter" | <i>H</i> Guard ring |
| <i>C</i> Thermometer giving temperature of plaster side of wall | <i>I</i> Blanket insulation of balsam wool |
| <i>D</i> Standard wall section | <i>J</i> Back and bottom of apparatus (wood) |
| <i>E</i> Thermometer giving temperature of outside of wall | <i>L</i> Hinged cover |
| <i>F</i> Second heat meter | <i>M</i> Clamp, to insure good thermal contact between parts |

FIG. 1. CROSS-SECTION OF APPARATUS FOR THE DETERMINATION OF THE THERMAL CONDUCTIVITY OF WALLS, FLOORS

and with its "wire side" next to the wall section which is then set in place as shown. The wall section *D* is surrounded by a guard ring—heating unit—which compensates for the heat which would otherwise be lost; thus the edge loss is supplied by an auxiliary heating unit instead of by the wall panel. The wall sections tested are made up in the form of a full scale panel 16 in. high and 23½ in. long, two wall studs (2 x 4 in., 16 in. on centers) being thus included in the frame sections.

A plaster thermometer, molded to fit the particular wall undergoing test, is placed snugly up against the wall. The resistance wire of this thermometer is

embedded in the surface next to the wall, so no appreciable temperature drop exists from the wall itself.

This is followed up by the second heat meter, and that by the waterback. The whole assembly is held tightly together by clamps as shown.

Insulite, $\frac{1}{2}$ in. thick and as large as the wall section being tested is used as the standard thermal resistance of the heat meters. On each side of this is securely fastened a flat resistance thermometer—identical in construction with the one already described—with the wires pressed into sure contact with the material. The entire heat meter is protected (from moisture absorption, etc.,) by a covering of rubberized sheeting.

Leads are brought out from all the thermometers to a slide-wire wheatstone bridge calibrated to read the temperature directly in degrees fahr. The thermometers are recalibrated regularly to insure the accuracy of the results. Connection is made to the desired thermometer by means of a mercury trough and cup switch. The resistance thermometers were made large to give an average value of the temperature over a considerable area.

Operation

The apparatus is assembled in the afternoon. The heating current in the main heater is adjusted to provide a temperature drop of about 50 deg. fahr. across the wall section. The current in guard ring heater is adjusted to a value that has been previously found approximately to offset the edge losses. The cooling water is turned on and the apparatus is then left to approach its equilibrium state over night.

Beginning the next morning at eight o'clock observations of the six resistance thermometers are made at intervals of from 15 min. to half an hour. These readings are continued throughout the day. In general, by 4 o'clock sufficient and satisfactory data has been obtained, and a new section may be put in place to be tested on the following day. Occasionally—due to a not wholly satisfactory adjustment of the heating current in the guard ring or for some other cause—the test is carried on over to the next day.

Data

As an illustration of the determination of thermal conductivities by this method, the following data is presented:

Description	B.t.u. per hour per sq. ft. per deg. fahr.	
	K	
"Standard wall".....	K	
5½ in. drop siding (cedar) tapered ⅝ in. thick, 3⅞ in. exposed.....	...	
Sheathing paper—Neponset building paper, 90/M.....	...	
Hemlock sheathing (ship lap) 7 x 13/16 in.....	...	
Douglas fir studding, 2 x 4 in. (1.5 x 3.5 in.) on centers actual.....	...	
Hemlock lath, 7/16 x 1⅜ in., spaced 1/4 in.....	...	
Gypsum plaster (well dried) thickness of lath + plaster = ⅝ in.	conductivity = 0.267	
Same construction as standard wall except that the lath is replaced by "Insulite," the plaster being applied directly to the Insulite.....	0.207	
Standard wall with Johns-Mansville "Acme Hairfelt" fastened between studding.....	0.200	
Standard wall with 1/8 in. "Flaxlinum" fastened between studding with cleats.....	0.192	
Standard wall with "Cabot's Quilt" ("Double ply") fastened between studding with cleats.....	0.190	
Standard wall with "Balsam Wool" fastened between studding with cleats.....	0.179	

Although the discussion has been confined to a consideration of wall sections, the method outlined is equally applicable to other elements of building construction and has already been so used with equal satisfaction.

DISCUSSION

P. NICHOLLS: The weakness of this article is that it fails to give proof of the accuracy that they get with the apparatus used. For instance, heat is stated to be measured by heat meters at each side of the apparatus and yet no data is given showing how the heat meters themselves were calibrated, or how far the accuracy of the values is known.

In the general problem of heat transmission, there are so many values that are obtained by different methods that if one desires to compare them and an author desires to have weight given to his value in preference to other values, then it becomes an absolute necessity that the presentation of his problem should contain all possible checks as to the percentage of accuracy that he gets and as far as he omits to do that, then, in future the value he has obtained will be given very little weight.

W. B. CLARKSON: I notice that their stated coefficient of "Insulite," as compared with "Flaxlinum" shows "Flaxlinum" to have the lower coefficient and under a different set of circumstances that might be reversed.

M. S. VAN DUSEN: Of course, this table of data does not show very much because it does not give the thicknesses of any of these materials except the one that is put in between the frame wall. Just saying, balsam wool or Acme hair felt, does not tell the thickness of the material. The hair felt might be twice as thick as the wool, so that comparison doesn't mean anything at all.

This method will work all right if it is applied to a solid wall containing no air space. The wall on which it is actually applied is a frame wall in which there is considerable convective circulation of air so that it is difficult to tell what the results will be under other circumstances, especially when extra heat is applied around the edges.

In a frame construction the height of the air space is vastly different, and the effective heat transfer will be different. In these experiments the object has been merely to compare the same wall with different insulating materials placed within, but it is quite possible that the effect of placing some material within the air space and dividing it into two parts would be considerably different with the air space only 2 ft. high than when it is 8 or 10 ft. high.

H. W. BROOKS: The engineer, in considering data of this sort will consider the fact that he has wind pressure, moisture and rain to contend with and will add a factor of safety.

In an electric circuit the contact resistance would be greater than the resistance of the circuit itself, that is, in a short circuit. There might be wide variations of contact pressure and thermal resistance between the plate and the medium to be measured. Some method of determining the pressure of heat plate and medium would have to be determined accurately and the same pressure maintained to get consistent results.

No. 682

DETERMINING THE EFFICIENCY OF AIR CLEANERS

By A. M. GOODLOE,¹ NEW YORK, N. Y.

MEMBER

THE difficulties encountered in determining the efficiency of air cleaners in terms of *weight* are well known. It is essential that the apparatus and method used in making such tests should be of such a nature as to eliminate as far as possible the natural sources of errors including those due to the human element. The apparatus and method described herein are used for determining by weight the efficiency of the average commercial air cleaner and have been devised primarily for the purpose of making a forced efficiency test of air cleaners in a laboratory or shop. It seems practical, however, for other dust determinations in the field.

In making an efficiency test of air cleaners under normal dust conditions, it is necessary to consider only the apparatus and method for obtaining by weight the correct amount of dust in a given quantity of air before and after it has passed through an air cleaner. Forced efficiency tests made where excessive artificial dust is introduced in air require considerable thought on the concentration and distribution of the dust, which should be such as to reproduce as near as possible a normal dust condition.

It is usually noted that the efficiency of air cleaning equipment varies greatly depending upon the dust concentration. This should not necessarily be the case provided all the dust particles are of the same size and weight. For instance, when there is a great increase in the dust concentration this is in most cases a result of the air containing particles of a larger size which would not normally be in suspension. The velocity and direction of the wind and peculiar air currents are factors that may cause a variation in the dust concentration. Therefore for all intents and purposes in the average case it is not far wrong to state that the efficiency of air cleaner will vary according to the dust concentration and for this reason more dependable and comparable results can be expected if the dust concentration and distribution are under control.

The amount of dust in the air should approach as nearly as possible the normal concentration which will permit a forced efficiency test to be made in a reasonable time, say from 1 to 6 hours, depending, of course, upon the efficiency of the air

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cleaner. An efficiency test can be made in this time by this method with a dust concentration of 5 to 10 grains per 1000 cu. ft., even when the efficiency of the air cleaner is 99 per cent.

During the forced efficiency tests made on air cleaners, artificial dust was introduced in such a way as to cause the dust to be diffused and distributed in the air in a way approaching as nearly as possible the manner in which the same quantity of dust would appear under normal conditions, and it consisted only of such size particles as would be carried in the air at a low velocity.

It may not be difficult to introduce in a duct the correct amount by weight of artificial dust in order to obtain the concentration desired. But dust in bulk introduced in a duct would, no doubt, contain large solid particles, and also large particles made up of a great number of small particles which have coalesced. These would be easily carried along in a high velocity air current, but would not normally be in suspension in the air. Therefore, it seems logical that the method

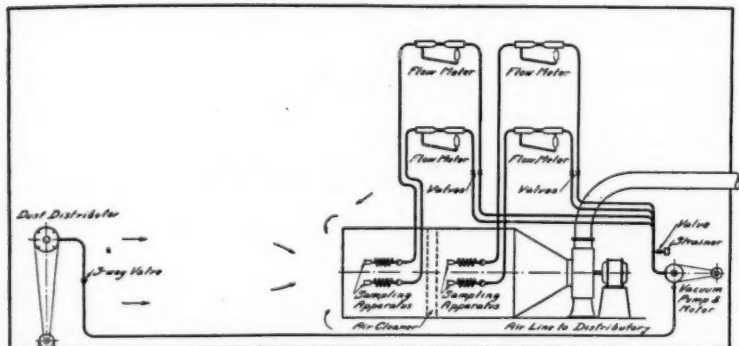


FIG. 1. GENERAL ARRANGEMENT OF APPARATUS

of distribution should be such as to eliminate particles of this size. This was accomplished by creating and maintaining a specific dust concentration in a large chamber or room and the air containing the dust in suspension was drawn from this chamber through the air cleaner, as shown in Fig. 1.

In the tests vacuum cleaner dust was used which was composed of dust particles previously floating in the air and settling on floors and walls, and it also consisted of soot, ash and the indescribable street dust. It is this kind of dust that creates the need for air cleaning equipment in a majority of cases. For this reason it appears to be quite representative. No doubt after this kind of dust has been collected in bulk there will be a coalition of the particles but probably this will be true to a certain extent with dust of any nature. However, by proper preparation and distribution, vacuum cleaner dust can be diffused in the air passing through an air cleaner in a manner approaching very nearly the condition in which it would appear under natural conditions.

A comparatively homogeneous distribution of dust can be maintained in a dust chamber by means of an apparatus known as a dust distributor (see Fig. 2), consisting of a steel drum α , 12 in. in diameter and 8 in. long. Inside the drum there are eight fins equally spaced b , 2 in. wide, extending the entire length of the drum.

The drum revolves on a hollow shaft *c* at approximately 100 r.p.m. A plug *d* is inserted midway in the hollow shaft. A nozzle *e* attached to a nipple is connected to the hollow shaft on the incoming air side. Three air outlet holes *f*, $\frac{1}{4}$ in. in diameter are placed in the shaft. In the incoming air line *g* there is located a three way regulating valve *h*. When the drum revolves the fins carry the dust up to a certain point where it drops to the bottom of the drum and causes a dust cloud. The plug in the hollow shaft causes the incoming air to be forced out through the nozzle which helps also to create a dust cloud in the drum.

Dust from the drum passes out through the three holes in the outlet side of the hollow shaft which are located on the underside of the shaft to prevent any large particles from being forced out. When the distributor is in operation the quantity of dust emitted can be controlled by the three-way regulation valve.

Viscous Impinger Used to Collect Dust

The apparatus for obtaining samples of dust is called a viscous impinger. There are several principles, not new but known to be effective, embodied in its design.

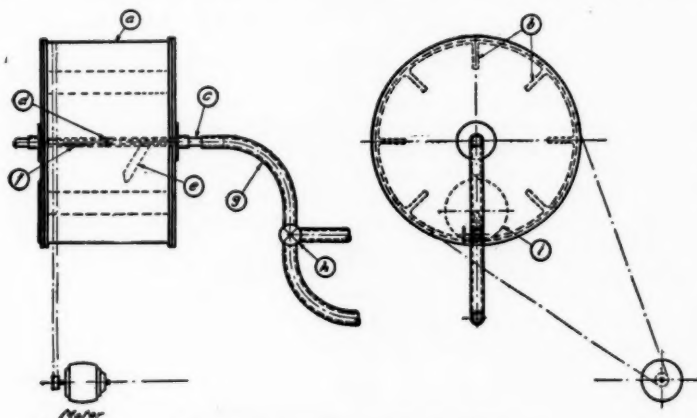


FIG. 2. EQUIPMENT FOR EMITTING DUST

As the air passes through the viscous impinger at a high velocity the dust particles are impinged against the vaseline coated surface by centrifugal action and abrupt changes in direction of the air stream. The viscous impinger, Fig. 3, consists of air intake *a*, spiral-like body *b*, velocity reduction chamber *c*, outlet connection *d*. The viscous impinger intakes are made with any desired diameter. The spiral body is made of glass tubing $\frac{5}{32}$ in. inside diameter. The overall length of the spiral body is approximately $1\frac{1}{2}$ in. The velocity reduction chamber is blown in the form of a hollow sphere $\frac{3}{4}$ in. diameter. The outlet connection is approximately 1 in. long, made of a tube with $\frac{5}{32}$ in. inside diameter. However this can be made of any suitable size.

Method of Using Impinger

To obtain a true sample of dust in an air duct it is essential that the air enters the sampling apparatus at the same velocity as the air in the duct. Viscous im-

pinger with intakes of the proper diameter can be selected, so that the entrance velocity will be nearly correct. A closer adjustment can be made by regulating the quantity of air passing through the sampling apparatus. A viscous impinger is prepared for use by warming it over an alcohol torch or by other means to about 100 deg. fahr. Vaseline used in the impinger is warmed to a fluid state. By



FIG. 3. SAMPLING APPARATUS

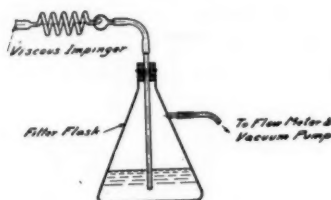


FIG. 4. AIR FROM IMPINGER PASSES THROUGH PHENOLPHTHALEIN SOLUTION

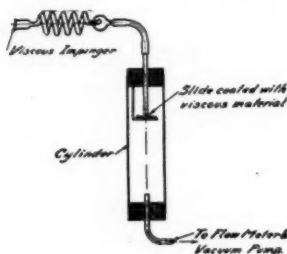


FIG. 5. AIR FROM IMPINGER STRIKES COATED SLIDE

means of a glass tube, just enough of the vaseline can be taken up to coat the inside walls of the spiral body of the viscous impinger. The liquid vaseline is inserted in the intake, then the viscous impinger is turned slowly till the vaseline coats the inside walls of the spiral body to within one half turn of the reduction velocity chamber. If it is found that too much vaseline has been inserted the viscous impinger is turned in the other direction till the surplus comes to the intake where it can be removed with a piece of cotton on a stick.

The coating will solidify within a few minutes, after which the viscous impinger is thoroughly cleaned on the outside, then weighed and wrapped, so that it is ready for use. Two or more viscous impingers are prepared in this manner for obtaining samples of dust in the air before and after it has passed through an air cleaner. When a sufficient amount of dust has been collected the viscous impingers

are again accurately weighed and the gain in weight will be equal to the amount of dust collected from which the efficiency can be determined in the usual manner.

About 0.5 cu. ft. of air per minute is drawn through each viscous impinger by means of a small motor driven vacuum pump. The quantity of air passing through the viscous impinger is indicated on a differential flow meter accurately calibrated by means of a gas prover.

The dust collected by the viscous impingers will not cause an appreciable reduction in the area of the air passage nor are there any other factors that will cause the resistance of the viscous impinger to change while making a test. Therefore no regulation of the air flow is necessary during the test run.

The average weight of the viscous impinger is 25 grams. When artificial dust is introduced and the concentration is from 5 to 10 grains per 1000 cu. ft. per min., an efficiency test can be made in 1 to 6 hours, depending upon the air cleaner efficiency and the nature of the artificial dust. It has been found by actual test that repeated results can be obtained without any appreciable variation, provided the conditions under which the tests are made are the same.

Hydroscopic Properties of Impinger

A great number of tests were made on the viscous impinger to establish its hydroscopic characteristics. A series of tests were made on the glass impinger without coating it with vaseline, and it was found that it neither absorbed nor gave up sufficient moisture to affect its weight. This can probably be attributed to the fact that the surface involved is comparatively small and that any variation in the surface film of moisture under the normal humidity and temperature changes is not sufficient to cause a noticeable error.

A further series of tests were run with the viscous impinger charged with vaseline. The fact was established that vaseline in this small quantity neither absorbed nor gave up sufficient moisture to affect the weight of the impinger, and there was no evaporation of the vaseline that could be detected in the length of time it required to make the test. The factor of dust was eliminated in these tests by using thoroughly cleaned air.

Efficiency of Impinger

After being satisfied that the viscous impinger could be handled without introducing serious errors the next step was to establish some facts regarding its efficiency. So far it has been impossible to determine the exact efficiency of the impinger because of the lack of time. However several tests were made to establish the dust removing qualities of the impinger in the following way:

A solution of phenolphthalein was placed in a filter flask. The impinger was connected as shown in Fig. 4, so that the air drawn through the impinger would be forced through the phenolphthalein solution. Caustic soda was put into a very fine powdered state with which a dust cloud was created near the viscous impinger; 36 mg. of caustic soda was collected by the impinger without the phenolphthalein becoming colored. The filter flask was again partly filled with water and a cloud of soluble prussian blue was created near the impinger; 41 mg. of blue was collected by the impinger without coloring the water in the filter flask. Further tests were made where the air was drawn through the viscous impinger from a large room, which contained a low concentration of vacuum cleaner dust. The impinger was connected to a cylinder holding a slide coated with a viscous substance, as in Fig. 5, and the appara-

tus was arranged so that the air from the impinger would strike the slide at a high velocity. The slide was first covered with Canada balsam and then with vaseline.

The results of these tests showed no accumulation of dust on the slides, and only slightly discolored spots. About 30 mg. of dust was collected by the viscous impinger in each test.

DISCUSSION

MARGARET INGELS: Even a glass weighing bottle, where there is nothing but a glass surface, will vary in weight. When moisture conditions are extreme the difference in weight of the bottle on different days often is .001 grams. This error in weighing, when the bottle contains a small amount of dust sample, may mean a large percent from the correct value of dust collected. Any method that measures dusts by weights should have a correction factor for the moisture if the calculations are to be at all close. This method is comparable to all methods for measuring dusts by weights. The principal one of this kind is the thimble, where the moisture makes the greatest difference but a correction is made for this moisture.

In Mr. Goodloe's method of measuring dusts he admits some air gets through, that is, the measuring apparatus is not high in efficiency. It is possible that the commercial apparatus will be as efficient as the instrument used to test it, but still be low on actual efficiency.

E. D. PRATT: When it comes to using the glass spiral for figuring efficiencies in air filters, it is well to consider Miss Ingels remark that nothing is perfect. There is, undoubtedly, some dust that gets through anyway, and, moreover, anyone who has ever made exploration of air velocity in any duct, square or circular will be surprised to find how the air velocity varies over any considerable cross section area. In other words, provision should be made in this duct to explore the velocity everywhere over the filter which has a considerable area, about 4 sq. ft.

Care should be taken to regulate the exhaust through the glass spiral, so that the air velocity or volume that goes through this spiral is representative of the air velocity that is generally in the filter.

Further, the effect of humidity must be considered. This is a method where you figure the efficiency from increase or the weight of the dust picked up. In handling air that we breathe the test should be carried on for a considerable time to obtain accurate values of the filter's efficiency.

W. H. CARRIER: The view that should be taken of this test is that it is a practical means of measuring rather than a strictly, scientific method. It would be interesting to know the amount of dust that did get through. For instance, what per cent of the total under the test conditions described?

F. R. STILL: From a manufacturer's standpoint, what we want is to have some kind of a standard apparatus that gives a qualifying ratio of what cleaning is being done. Some specifications for air cleaning apparatus claim that 90 per cent of the dust can be removed. Is it a question weight or quantity. There are dozens of different kinds of apparatus on the market for dust determina-

tion. All of them give different results and it takes a great deal of skill with most of them to determine what is the amount of dust. If a machine can be developed that will establish a standard for quantity of obstruction through a filter in a given length of time, then we will have a machine of commercial value and one that will ultimately give a ratio of the amount of dust that can be eliminated.

H. W. BROOKS: One of the interesting features is the bell-shaped mouth of the viscous impinger. I would like to ask Mr. Goodloe if he feels that there is any possibility of any currents in the air producing appreciable error in the results.

A. M. GOODLOE: I always considered that when the air was once inside of the entrance of the intake, any turbined effect in its passage on through would not effect the results, because the bell-shaped impinger is coated with viscoscene and the dust is caught up. In order for that dust to get in the intake, air has to get inside and pass through, which seems an impossibility. I have found in my experiments that where I reproduce the same conditions that my results will not vary over one or two per cent, and I can, furthermore, reproduce the same results in a few per cents. I have tried some of the drying methods and I got 115 per cent one time and the next time 25 per cent, but in this apparatus I get repeated results.

MISS INGELS: One of the principle faults with all the 40 or 50 dust determinators on the market is that it takes a skilled person to get the results after they have been obtained.

DR. E. VERNON HILL: If a filter is put on the market that is 97 per cent efficient considering that you can handle possibly 100,000 ft. of air, one per cent lack of efficiency would mean you would have 1000 cu. ft. of air going through that was not clean, and when you consider that this 1000 cu. ft. of air is going through hour after hour and day after day, why the efficiency of equipment is very low. It is that low efficiency, that difference in efficiency between the various filters, between filters and washers, that must be determined. The measurements are very important.

W. H. CARRIER: Probably Dr. Hill misunderstood me when I said 1 or 2 per cent, and I should like to correct this impression. I didn't mean 1 or 2 per cent difference but per cent of the quantity, your final dust determination.

P. NICHOLLS: The general statement made by Mr. Goodloe about the weights and relative differences in weight for moisture and changes in moisture or change in weight in the viscoscene, does not give any statement as to the amount of moisture that is being passed through. For instance, in using such an instrument on the clean side of an air washer, where moisture might be very considerable it seems to me that the plans of the spiral tube and the coating inside is about as ideal a collector of dust that can be imagined and has all the simplicity. It has, however, a disadvantage that the weight of the instrument itself is considerable, and to get an appreciable accuracy with the addition of the small amount of dust, if the air is comparatively clean, the change in weight will be comparatively small, and, therefore, it is absolutely essential, in order to accept the accuracy claim, that very definite figures should be given, and, therefore, definite, repeated and well-designed tests be made to show that other factors do not influence the weight he gets from it.

H. W. BROOKS: The device used depends to a certain extent on the principles that are used in steam separation, of which there are in general two: one which involves a change in direction and flow; the other a principle of the receiver separator. It seems to me that a somewhat larger receiver should be used at the point

where the square tube ends. I believe that there would be a certain precipitation in a larger capacity in decreasing the velocity of flow and thus get a deposit of the final dust. It has the disadvantage that it increases the weight of the apparatus.

A. M. GOODLOE: No regulation of air flow is required during the tests. The dust deposits in such a manner that it does not restrict materially enough to effect the air flow, and I am always informed of the amount of air that goes through the impinger.

No. 683

AIR HANDLING AND HUMIDITY PROBLEMS IN WISCONSIN PAPER MILL

By ARTHUR T. NORTH,¹ NEW YORK, N. Y.

NON-MEMBER

THE plant of the Appleton Coated Paper Co., located at Appleton, Wis., is of considerable size, having a production capacity of 50,000 lb. of coated paper every 24 hours with the five coating machines in use early in 1923. Three additional machines will be installed when the new buildings, now under construction, are completed. The plant is operated continuously from 1:00 A.M. Monday until 6:00 A.M. Sunday each week.

The principal thing of interest for the heating and ventilating engineer, in this plant, Fig. 1, is the method used for drying the paper and heating the building. A brief explanation of the coating and drying process may enable the reader to understand better the description of the heating and drying methods given later. The plain paper is not made in the mill but is delivered in large rolls similar to the news-print paper rolls with which we are all familiar. The weight of paper and width of rolls varies according to the production requirements.

The rolls of paper are mounted in front of the coating machines, Fig. 3, and the paper passes through them where the coating mixture is applied and evenly distributed over the surface. The coating mixture is a thick substance made of clays, pigments, casein and other ingredients. As this flows quite freely, it naturally contains a considerable amount of water which must be evaporated to a certain degree of dryness. The speed at which the paper passes through the machine is necessarily quite slow.

After the paper receives its coating, it travels a sufficient distance horizontally for the coating to harden enough to retain its place and not run. The paper is then elevated to a height which will permit the forming of loops about 8 ft. high and about 6 in. apart at the top of the loop. These loops which can be seen at right, Fig. 2, move forward automatically through the drying zone as new loops are formed. As the drying zone is about 60 ft. wide and 140 ft. long there is probably about 4500 ft. of paper passing through the zone between a coating machine at one end and the winding machines at the other end, where the coated paper is re-wound into rolls. These rolls of coated paper are then taken to the calendering machines through which the paper is passed between rolls under heavy pressure,

¹ Engineering and Associate Editor of *American Architect*.

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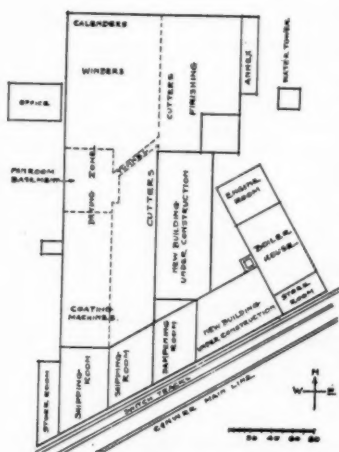


FIG. 1. PLAN OF MILL

thus bringing the product to a uniform thickness and finished surface. The paper is then cut to size, sorted, counted, and packed ready to be sent to the shipping room or warehouse.

The length of the room is about 315 ft. The height of the outside wall is about 15 ft. above the floor, and the height of the underside of the roof at the ridge is 30 ft. Along the ridge of this building, over the drying zone are ventilating monitors which are now closed tight against the outside atmosphere. Hot-air ducts are placed in this part of the building, extending from the coating machines to the calendering department, Fig. 5, decreasing in size towards the end. These ducts are over each of the five lines of paper and have openings close together in the underside from which the hot-air is forced down into the loops of drying coated

paper. After serving its purpose in drying the coated paper this air heats the entire plant except some small portions which will be mentioned later.

There is a basement under the one story cutting and finishing space in the north-east corner of the building. On the first floor, over this basement are placed openings with grates through which the air is drawn into the basement, then through the tunnel leading to the fan room. The entire mill proper is in one open room where the paper is coated, dried, wound into rolls, calendered, cut, sorted and packed. In this room temperatures and humidities are maintained as follows: at coating machines, 76 deg. fahr., humidity 83 per cent; drying zone 110 deg. fahr.; winder floor space, 79 deg. fahr., humidity 68 per cent; calender space 80 deg. fahr., humidity 76 per cent; finishing space 72 deg. fahr., humidity 78 per cent. The proper humidity at the drying zone has been determined by experimentation and it is not permitted to be here disclosed.

The temperatures outside of the drying zone are controlled by manipulating the dampers in the outlets in the floor. By this means the flow of the air is controlled at will. This control is not automatically regulated but is done by hand as required. The humidity is regulated by the admission of fresh outside air into the fan room, in such quantities as may be required. No artificial means of humidifying the air by water, steam or vapor is used. It is done entirely by controlling the admission of outside air. It should be noted that the air in the mill is continuously recirculated which makes a saving in steam consumption in the heating coils. The air is always fresh and extremely agreeable in every part of the mill.

The heating apparatus consists of 2508 sq. ft. of radiation composed of 1 1/4 and 2 in. wrought iron pipe, except that 314 sq. ft. of cast iron radiation is installed in the office. The balance of this radiation is used for the sole purpose of keeping the mill warm during the time the operation is suspended over Sunday. Three stacks of heating coils containing 8040 sq. ft. of radiating surface are installed in the fan room. Three 72 in. Sturtevant fans are located in this room, one of which

is a reserve unit. In this room the air is conditioned as previously described and discharged into the hot-air ducts at a temperature of 160 deg. fahr.

The entire plant contains, aside from the boiler house and the engine room, 1,464,755 cu. ft., with 436 sq. ft. of exposed wall and 4115 sq. ft. of glass. The heating, drying and ventilating is accomplished entirely by exhaust steam of which the average consumption for the month of December, 1922, consisted of 167,778 lb. of steam per 24 hours as measured by a Republic flow meter. The entire steam production averaged 207,247 lb. per 24 hours, 39,466 lb. of which is used in process work, boiler feed pumps, and fan engine for forced draft, and which does not pass through the flow meter and the engine. The average coal consumption per 24-hour day during this period was 20,361 lb., and evaporation of 10.177 lb. of



FIG. 2. FINISHING ROOM—NOTE PAPER IN FOLDS AT RIGHT

steam per pound of coal. The boiler plant, Fig. 3, consists of one 500-hp. Kidwell water tube boiler and one 500 hp. Babcock & Wilcox water tube boiler equipped with Type E. stokers. Only one boiler is in service at a time. The power plant consists of one 20 x 32 Nordberg poppet valve engine and one General Electric 460 kw. generator. All of the power in the plant is derived from individual motors on each machine.

The back pressure on the engine as indicated by a gage never exceeds 1 lb. as a relief valve is set to operate at that pressure. The condensation from this entire system is taken from the coils and stacks in the fan room by a 2 in. rotary pump which is manufactured by the Hayton Pump and Blower Co. A compound gage placed in the suction of the pump shows from no pressure to $\frac{1}{4}$ lb. pressure. A thermometer placed approximately 4 ft. above the pump in the discharge line shows a temperature of the return water which varies, with the load on the engine, from 200 deg. to 220 deg. fahr. The feed water is returned to the boilers at a temperature of 210 deg. fahr.

The unique feature of this system is in the method of conditioning and handling the air. There is no steam evaporated for humidifying, and the humidity in the drying zone is under complete control and can be brought to any desired percentage by manipulating dampers in the ducts entering the fan room, through which air is brought from the different parts of the mill and from out of doors.

When it is realized that the entire mill, except that part adjacent to the railroad tracks, is in one room and that there are five different temperatures and humidities maintained therein from one source of supply, then it is easy to understand that the results accomplished are out of the ordinary. The entire plant is heated, and drying is accomplished entirely by the use of exhaust steam, except that live steam is used for direct radiation heating on Sunday when operations are suspended. Another unusual feature of this system is that there are no vacuum or steam traps at any place in the system, the exhaust steam leaves the engine and makes the complete cycle returning to the feed water heater by the operation of one pump

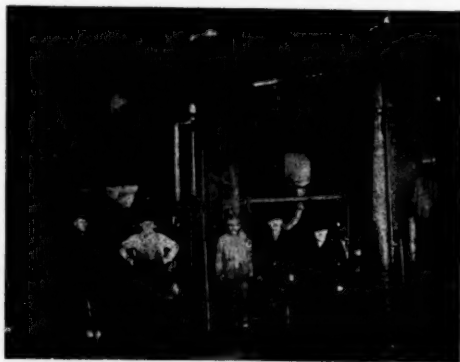


FIG. 3. SCENE IN BOILER ROOM

and the air from the entire system is discharged to the atmosphere through a $\frac{3}{4}$ in. pipe from the feed water heater.

In the summer time, when heat is not needed, the drying zone is kept at the standard temperature and humidity, and the remainder of the mill is cooled simply by opening the windows which allows fresh air to be drawn in and drawn through the tunnel into the fan room. By means of the rapid circulation of the air, the employees work in comfort during the warmest weather with a temperature not exceeding that of the outside notwithstanding that air is being discharged from the fans at a temperature of 180 deg. fahr. During extreme cold weather all of the exhaust steam is condensed in the heating coils and stacks.

These unusual features were devised by the chief engineer of the plant, J. C. Stillman, for some of which patents are pending. To Mr. Stillman and Benjamin Vaughn of the Hayton Pump and Blower Co., the author is indebted for the data here given and for several opportunities to inspect the plant thoroughly.

No. 684

HEAT EMISSION FROM HEATING SURFACES OF FURNACE

By A. P. KRATZ,¹ URBANA, ILL.

NON-MEMBER

A KNOWLEDGE of the amount of heat emission from the heating surface of a warm air furnace, and of the relative value of the different parts of the heating surface is essential in fixing a basis for the intelligent design of such furnaces. In order to obtain basic data of this kind, a study has been made of several series of warm air furnace tests conducted under varying conditions. The tests were all made with anthracite coal on the plant shown in Fig. 1 and formed part of the program of the investigation of warm-air furnaces conducted by the Engineering Experiment Station at the University of Illinois under a cooperative agreement with the *National Warm Air Heating and Ventilating Association*. The investigation is being conducted under the general direction of A. C. Willard, professor of Heating and Ventilation and head of the Department of Mechanical Engineering.

For the analysis given in the present paper only the tests in which the surface temperatures for the different sections of the heating surfaces had been observed were selected. All of these tests were made on the same castings and included tests made with the following diameters and types of casings: (1) 52 in. with no inner lining, (2) 52 in. with black iron lining spaced 1 in. from the casing, (3) same as (2) with the addition of a radiation shield, (4) 50 in. with no inner lining, (5) 50 in. with inner lining and 1 in. air space, (6) 56 in. with inner lining and 1 in. air space, (7) 56 in. with inner lining of corrugated tin and asbestos paper placed against the outer casing, (8) 52 in. with inner lining and 1 in. air space, and vertical bonnet having side outlets. For all casings except (8) a conical bonnet was used.

The results were calculated on the basis that the heat emitted from the heating surfaces consisted of the total heat developed by the combustion of the fuel, minus the sum of the heat lost through the grates to the floor, the heat emitted by the front and that which escaped with the flue gases. From the heat thus calculated, and the observed temperatures of the heating surfaces, it was then possible to calculate the heat emitted by the separate sections of the castings.

The computed quantities of heat emitted per square foot of surface per hour were plotted against the combustion rate. The combustion rate was selected as the base for plotting because it is a factor that may be readily obtained or logically assumed in practice.

¹ Research Professor in Mechanical Engineering, University of Illinois.
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Discussion of Results

The curve in Fig. 2 gives the mean heat emitted per square foot per hour for the total heating surface of the castings. The fact that the points for all types of casings used fall on a smooth curve indicates that the total heat emission from the castings is independent of the type of casing or bonnet. This curve is useful in designing other furnaces in which the distribution of the heating surface is the same

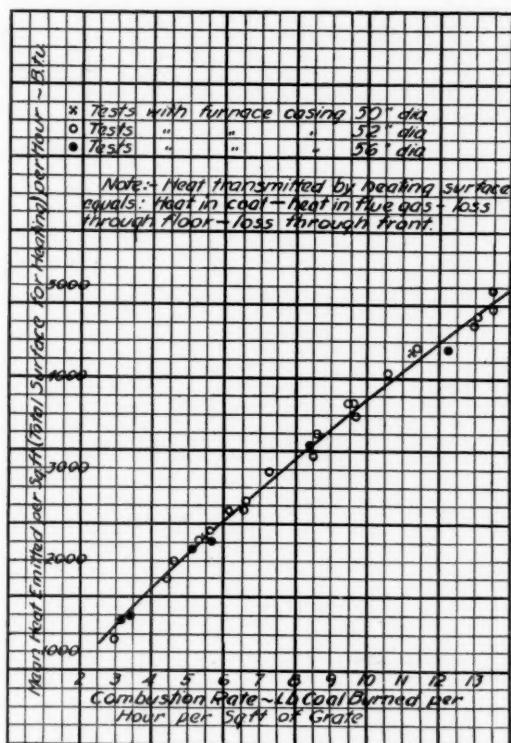


FIG. 2. CURVE SHOWING THE MEAN HEAT EMITTED PER SQUARE FOOT PER HOUR FOR TOTAL HEATING SURFACE OF CASTINGS (GRATE AREA 2.88 SQ. FT.)

as in the one tested, but it gives no indication of what might be expected in case the distribution was not the same.

The curves in Fig. 3 show the heat emitted per square foot per hour by the separate sections of the heating surfaces and may serve as a guide in designing and proportioning these surfaces for the circular-radiator type of furnace. In obtaining the curves no tests were used except those on the casings having the black iron linings with 1 in. air space, since these casings were proved to be the best and the

most practical. These curves also indicate that the heat emission is independent of the type of casing.

Table 2 has been compiled from data taken from the curves of Figs. 2 and 3 and gives the relative value of one square foot of heating surface for the different sections expressed both as a percentage of the mean heat emission from the total surface, and as a percentage of the heat emission from the surface of the firepot.

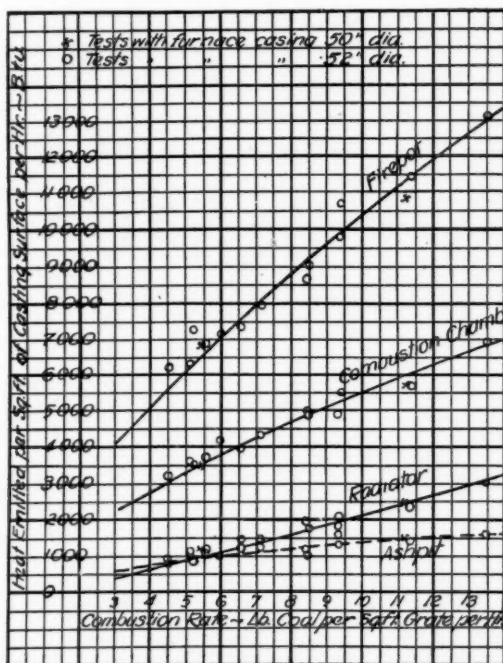


FIG. 3. CURVES SHOWING HEAT EMITTED PER SQUARE FOOT PER HOUR BY SEPARATE SECTIONS OF HEATING SURFACE (GRATE AREA 2.88 SQ. FT.)

From this it may be noted that one square foot of the combustion chamber surface emits approximately 53 per cent, one square foot in the radiator about 17 per cent, and one square foot in the ashpit about 15.1 per cent of the heat emitted by one square foot of the firepot surface.

It should be noted that the ratio of the heat emitted by one square foot of the combustion chamber to the heat emitted by one square foot of the firepot is not the same as the ratio of the total amount of heat emitted by the combustion chamber to the total amount emitted by the firepot. This is true because the areas of the two sections are not equal. Similar conditions exist for the other sections of the heating surface.

In the furnace tested, the firepot surface emitted approximately 30 per cent, the combustion chamber 40 per cent, the radiator 24 per cent and the ashpit 6 per cent of the total heat emitted per hour. All of the heat emitted from the heating surfaces was not absorbed as useful heating effect by the air circulated. The useful heating effect (heat in the air at the bonnet) varied from 82 per cent of the heat emitted for a 4 lb. combustion rate to 64 per cent at a 12 lb. combustion rate and had a mean value of about 75 per cent at a 7 lb. combustion rate.

TABLE 2. RELATIVE VALUE OF ONE SQUARE FOOT OF HEATING SURFACE

Section	3 Lb. Combustion Rate			7 Lb. Combustion Rate			12 Lb. Combustion Rate		
	B.t.u. given off per sq. ft. per hr.	Relative value of 1 sq. ft.		B.t.u. given off per sq. ft. per hr.	Relative value of 1 sq. ft.		B.t.u. given off per sq. ft. per hr.	Relative value of 1 sq. ft.	
		Per cent of mean for total surface	Per cent of firepot		Per cent of mean for total surface	Per cent of firepot		Per cent of mean for total surface	Per cent of firepot
Firepot	4000	310.2	100.0	7850	281.5	100.0	11,800	269.0	100.0
Combustion Chamber	2100	164.0	52.5	4200	150.5	53.5	6150	140.0	52.0
Radiator	500	39.1	12.5	1350	48.4	17.2	2700	61.5	22.9
Ashpit	600	46.9	15.0	1200	43.0	15.3	1600	36.5	13.6
Mean for Total Surface	1280	100.0	32.0	2790	100.0	35.5	4390	100.0	37.2

Conclusions

1. The heat emission per square foot of heating surface is independent of the type of casing or bonnet for a given set of castings.

2. In the furnace tested, approximately 30 per cent of the total heat emission was emitted by the firepot, 40 per cent by the combustion chamber, 24 per cent by the radiator and 6 per cent by the ashpit.

3. For the cast iron circular-radiator type of furnace, one square foot of surface in the firepot is equivalent to approximately 1.9 sq. ft. in the combustion chamber, 5.9 sq. ft. in the radiator, and 6.7 sq. ft. in the ashpit.

4. At a mean combustion rate of 7 lb. per sq. ft. of grate surface per hr., approximately 75 per cent of heat emitted by the heating surfaces appears as useful heat in the air circulated based on the heat in the air at the bonnet of the furnace.

DISCUSSION

D. R. RICHARDSON: There has been a great deal of harm done to the furnace business by the use of cast iron radiators in furnaces burning hard coal. It is easier to get 6 or 8 ft. of flame from soft coal than it is a few inches of flame from hard coal. It is a mistake to have cast iron radiators, except near the seashore or in damp cellars. The impossibility of heating up quickly in the morning with a furnace with a cast iron radiator, using hard coal has retarded the progress of the industry. Using the same furnace with a sheet iron radiator, however, reduces the heating up time from an hour or more to 15 or 20 minutes.

Another objectional factor is the size of casings on furnaces. When the case is close to the radiator it is death to the firebox. Most furnaces lack a proper supply of cold air and having a larger size casing allows a freer flow of air, and it not only preserves the firebox but saves the bodies and radiators.

E. B. LANGENBERG: In the work being done at the University of Illinois, we have been trying to get down to basic facts as affecting a warm air heating plant. We have come to the conclusion (and some of us have been in steam and hot-water work) that a residence can be heated properly with a warm-air system.

I asked Professor Willard some time ago why it was that during the war they said that a 24 in. firepot should have a 48 in. casing. Why not determine through research work what is the correct diameter of casings that will save a furnace and at the same time give us the volume and capacity of air inside? Experimenting with this, it was discovered that if a baffle plate was put inside the casing halfway between the casing and the furnace proper, the outside temperature of the casing can be reduced from 150 to 105 deg. That means a big saving in heat, because we have practically four surfaces for the air to come in contact with, and in handling air we must bring the air in contact with the heated surface before we can get it warm.

Last December 71 manufacturers met in Urbana, Ill., and contributed \$25,000 to establish a test house for the purpose of proving the facts that we are learning in the Laboratory of the University of Illinois and to show you and the public that the facts are true and will work out in practice. The test house is a typical house with eight rooms that will meet almost any condition. The construction is such that the exterior walls can be made of frame, brick veneer, stucco, hollow concrete blocks and tile. The present construction is frame. The results of this work in this particular house are going to be far-reaching in their effects on the public because we can conclusively say that our tests have proven certain things by actual practice.

No. 685

MEASURING HEAT TRANSMISSION IN BUILDING STRUCTURES AND A HEAT TRANSMISSION METER

By P. NICHOLLS,¹ PITTSBURGH, PA.

MEMBER

This paper deals with the measurement of the heat flow through walls, more particularly of existing structures. It outlines the principles employed and describes in detail work done at the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS' Research Laboratory in the attempt to develop a Heat Transmission Meter which will indicate instantaneous flows. It deals with the difficulties involved in such measurements and the variable factors in building materials that will influence the heat transmission constants. It closes with a short review of the present state of our knowledge, future requirements, and indications of the probable trend of investigational work.

THE heat transmission referred to in this paper is that through walls, roofs and floors or, expressed in more general terms, that in or out of flat surfaces.

There are several principles available for the measurement of the rate of flow of electric power, but that of heat is more circumscribed, and also not capable of the same precision. Heat flowing through a body does not, as far as our present knowledge goes, alter it because of the flow, and if there is any change, it is that produced by the actual or relative temperatures which result. Heat flowing into a body does produce changes which are a measure of the quantity of heat received—such as a change in volume, or a solid into a gas or liquid—but as soon as it flows through and the rate of loss equals that of gain, then these changes cease.

There are thus only three principles available, namely, absorbing after it has passed through the body, supplying a fixed amount, or directly or indirectly measuring the temperatures at two or more points, and from the known thermal properties of the material estimating the flow. To complete the explanation direct measurement means, for example, measuring the change in size at those points. If heat were flowing along a rod the temperature at any section would be given

¹ Research Head, A.S.H.-V.E. Laboratory. Copyright, 1924, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

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by the diameter, and an instrument which would indicate the diameters at two sections could, theoretically, be calibrated to show directly the rate of flow.

Absorbing the heat after it has passed through the body has some applications and has been used advantageously by experimenters. It has usually consisted in absorbing it in coils through which water is circulated or in a tank containing ice. In both methods the coils or tanks must be guarded against losing heat themselves, and perhaps this is more easily accomplished with the ice. They are both limited, if accuracy is desired, to the measurement of larger heat flows.

The supplying of a definite amount of heat has been the method usually employed by investigators, and the various types of laboratory apparatus have often been described and this need not be repeated.

There is one method which needs special mention as it has been used to measure the flow through the walls of buildings with natural outside weather conditions.

This method was first proposed in 1912, by A. H. Barker of England and used in tests carried out at the University College, London. Its principle is shown in

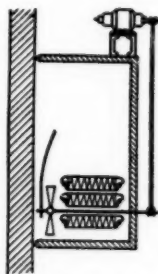


FIG. 1.

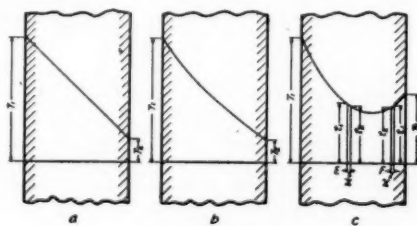


FIG. 2.



FIG. 3.

Fig. 1, and it consists of a box placed over a portion of the wall. This box has in it heating coils and a fan to circulate the air gently. The air inside the box is kept at the same temperature as that in the room by means of a thermostat which regulates the current in the heating coils. Presuming there is no heat transfer through the walls of the box, all the heat passing through the portion of the wall covered by the box is supplied by the coils, and by metering this and recording the various temperatures, a log can be plotted showing these values against time.

This method has been used recently to advantage by the Government Testing Institute of Sweden and the Norwegian Technical Institute of (Technology). In the Swedish work constant temperatures are maintained on both sides of the wall, and in the Norwegian small houses were constructed and the interiors maintained as far as possible at a constant temperature, but the natural weather conditions existed outside.

This method is the nearest approach to measuring the heat flow through a given area of a building wall and still maintaining natural conditions. The surface coefficient on the hot side will be a little disturbed, but this will be small compared with the total thermal resistance. It requires close regulation of the heat supplied in addition to the temperature measurements and needs either close attention or

delicate thermostatic control. The large surface area of the box would permit of an appreciable interchange of heat through it, but a further refinement could be added by dividing this area into any number of sections and equalizing the temperature on the two sides, by heating elements built into each section.

Measuring the whole heat input required to keep a building at constant temperature has also been employed, and by constructing special types, over-all results are obtained. Tests of this type have been made in Minnesota, and more exten-

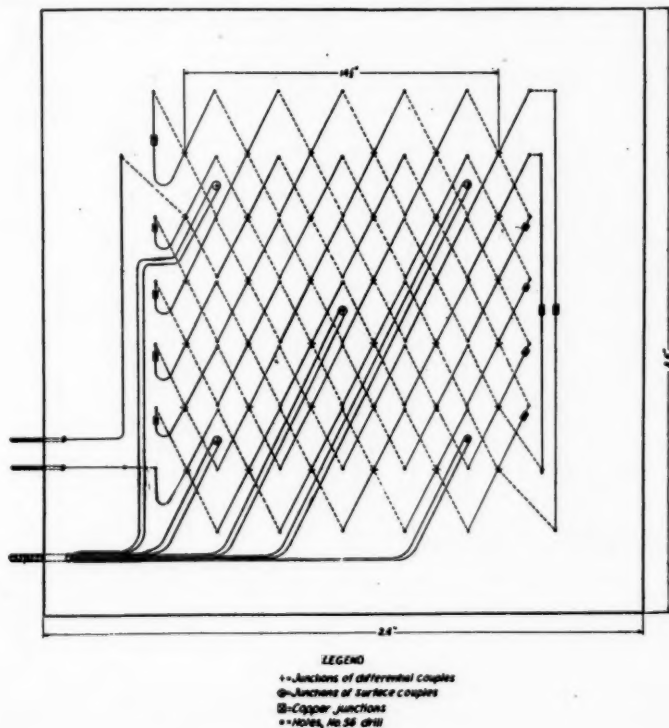


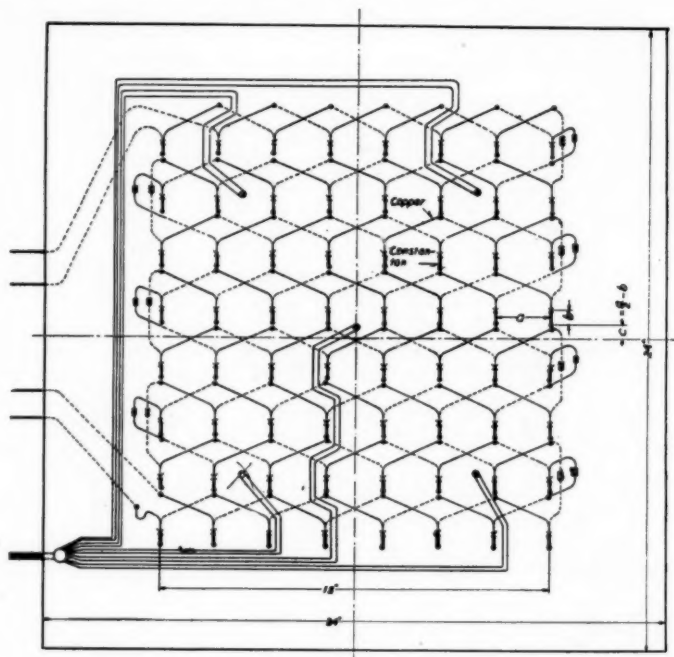
FIG. 4.

sively in the Norwegian tests, the overall values being supplemented by Barker's method applied to the window, roof and walls so as to separate the factors.

The method of measuring the temperature change along the path of the heat flow and from the known thermal resistance between two points calculating the rate of flow, has two possibilities.

Heat flow can be estimated from the surface transmission coefficient. As far as this is possible and accurate it is the simplest and causes least disturbance to the natural condition. Unfortunately the surface coefficient depends on several uncertain factors, the nature of the surface, its location both actual and relative, the temperature and motion of the air, and the temperature of surrounding bodies.

As it is difficult to specify what these are, and as the laws connecting them with the heat are largely empirical, it is hardly to be expected that any great accuracy can be obtained in this manner. It can, however, be used with fair accuracy in special instances when the flow is comparatively large. For instance, it has been shown that a very good measure of the heat dissipated from the outside of a covered steam pipe can be fairly well estimated by measuring the air and surface temperatures, provided the latter is done in a definite way.



LEGEND

- X = Junctions of differential couples
- = Junctions of surface couples
- = Copper junctions
- = Holes, No. 56 drill

FIG. 5.

Our knowledge of surface action has been enlarged by the data contained in Report No. 9 of the Food Investigation Board of England on the "Transmission of Heat by Radiation and Convection." Although no great accuracy could be obtained by using the natural surfaces, there is the possibility of setting up a slightly artificial condition, forming a definite surface and surroundings, and using fixed methods of temperature measurements, calibrating it by passing known heat flows.

The second possibility is the temperature gradient in the material. Since all materials have a resistance to heat flow by conduction, a flow cannot occur

without a fall in temperature along its path. Fig. 2 shows types of gradients that may exist in a uniform wall with T_1 and T_2 surface temperatures. With constant T_1 and T_2 and uniform conductivity it will be *a*. With constant temperature, but the conductivity varying with the temperature of the material in the form of the coefficient $= k(1 + aT)$, it will be *b*. In building walls the temperatures T_1 and T_2 are not constant but vary with more or less rapidity, and the curve for an outside wall at noon in the late autumn might be as in *c*. If the curve and conductivity coefficient be known the heat flow at any plane is given by $H = (t_1 - t_2) \frac{k}{x}$, x is the width of a thin layer, with conductivity k , and t_1 and t_2 the temperature of the faces of the layer. The heat must flow in the direction of the lower temperature, and thus in Fig. 3-c, is flowing away from T_1 at *E*, and towards it at *F*.

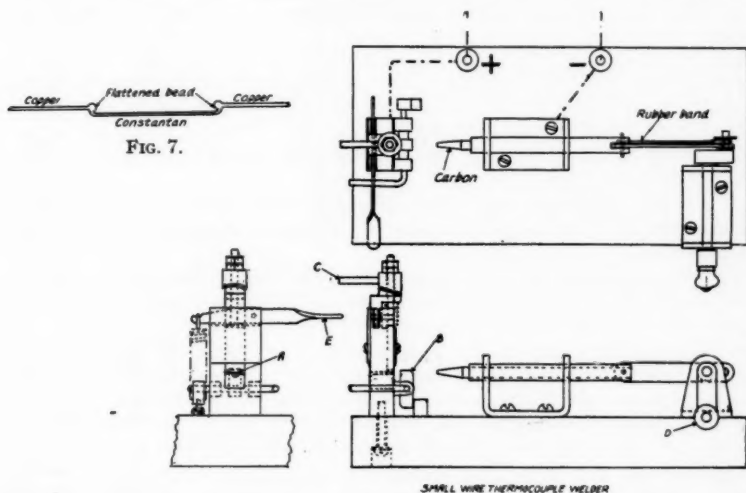


FIG. 6.

The variations inside the wall are of secondary interest, and the flow in or out of one of the surfaces the more important, such as inside surfaces of buildings in either hot or cold weather. There is little possibility of obtaining any reliable results from using the wall itself as the conductivity is too uncertain, and though difference in temperature between the surface and a small depth below can be approximated by the use of thermocouples, yet such readings are subject to large errors.

There is the option, however, of placing a layer of material over the surface whose conductivity is known and measuring the flow of heat by its temperature difference. If the thickness and thermal resistance of the layer are comparatively large, this can be done with presumed fair accuracy. Cork board of one inch and upward has been used, since this material has been tested more than any other material and its conductivity can be approximately predicted. This, however, adds considerable thermal resistance to the wall, and if thin layers are employed

the proportional errors in measuring the surface temperatures may be appreciable and, at best, leave the accuracy doubtful. Such surface temperatures would be measured with thermocouples, but, even then, since the rate of fall in temperature in the air layer next to it is very rapid, the couple will project enough to put its reading in error. The same error would occur if the temperature were measured by the change in resistance of a wire on the surface. To overcome this the wires must be cemented on so that they are invariable, and their temperature readings obtained by calibration tests in which known rates of heat flow are passed through them.

Such a plate when calibrated would be a Heat Transmission Meter since its temperature reading would directly give the rate of heat flow through it in whatever position it be used. Placing it on a wall will give the flow at that surface through the wall plus the plate, and as far as its thermal resistance is small compared with that of the wall, the flow through the part not covered can at least be approximately determined. It will, however, have the disadvantages of added resistance and changing the nature of the surface, but it will at least give a direct reading of the heat flow with the minimum of apparatus and without impressing a source of heat, can be closely estimated for instantaneous values, and more so for average values over a sufficient time.

The work done on developing such meter plates is described and is governed by the following principles:

1. If they were to be of any value sufficient evidence would have to be offered of the order of their accuracy to ensure the acceptance of values they would give in application.
2. The final plates must have the minimum thermal resistance consistent with the use of not too expensive auxiliary instruments, must be permanent in their calibration, and must be fairly rugged.

Construction of Plates. It was decided to make the plates two feet square. If it were found advisable to cover a larger area in their application, it could be done by using several similar plates and four of them would cover 16 square feet. Part of the plate must be the guard ring and the design of testing hot plate fixed the center area to be covered by the thermocouples as 15 x 15 inches.

Some doubt was felt on the possibility of obtaining consistent results with thin plates, as well as doubt as to the possibility of making them so that they would be permanent and not fail, so it was decided to make the first set of several materials of different thicknesses, as a comparison between their readings would help to better bring out the causes of variations and confirm their accuracy.

The next thing to decide was the method of measuring the temperatures. Even if the plates were thick it would be necessary to average it as far as possible over the whole surface, and more so if they were thin. There was much in favor of using the variation of electrical resistance method, as the wire would average over a larger portion of the surface, and also the construction would be cheaper. Against it was the need for a high class of instrument not usually available, and the great danger of change in the resistance due to strain when used on the plates, and to the temperature strains, as well as possible variations due to the cement holding them to the plates.

Thermocouples only register point temperatures and are expensive to apply but afford a means of multiplying the reading by using them in series, as well as having permanence within the temperature range of application. They were therefore adopted.

Thermocouples give a convenient means of measuring the difference in temperature of the surfaces by having alternate junctions on the two sides. This could be done by bringing the wire around the edges, or by taking them through holes in the plate. The former makes a very complicated wiring system and greatly increases the electrical resistance of the couple system.

In addition to measuring the temperature difference one surface temperature is needed in order to be able to connect the differential e.m.f. with some zero, and also because of possible variation of thermal conductivity with temperature.

Since the plates were to be calibrated by test, and not by the known conductivity of the material, there is no necessity to deal with actual temperatures but only with the readings given by the thermocouple system, and this in itself takes care of variations in the closeness of the wires to the surface, of the e.m.f. given by different junctions, and of the plate material, provided a given rate of heat flow at a given plate temperature always gives the same couple reading.

As the wires were to be taken through the board it is necessary to consider how this affects the readings given. Since, Fig. 3, if the temperature of surface AB is higher than that of CD , the much higher conductivity of the wire will conduct the heat along it faster than will the material and thus the temperature difference between the junctions will be less than T_1 and T_2 . As the maximum e.m.f. with the minimum electrical resistance is desired there is some length which will give the maximum accuracy. The mathematics of this is treated in Appendix A, and the useful result is developed that with copper-constantan couples the copper wires should be four times the length of the constantan which is very helpful in reducing the electrical resistance. Also that the wires could be made very short without an appreciable loss in efficiency. Since, however, the junctions taking their final temperature might bring in time, on the first set of boards the wires were made long, and those on the thinner board No. 40 B & S. The use of this fine wire also ensured that they could be cemented close to the surface and would project little above it.

The wiring diagram for the first set is shown in Fig. 4. It is good for any number of couples as long as they are the square of an even number.

The diagram adopted for standard boards is Fig. 5. This gives the couple points a uniform distribution while permitting any lengths of wires. All the thinner plates have the couple systems divided into two halves, from which leads are brought out so that they can be read in series or multiple, thus giving more flexibility in obtaining the most accurate and sensitive reading with various grades of potentiometer.

Five surface couples per plate were adopted, and these were mostly put in series. The multiple arrangement is however sufficiently accurate and less liable to breaks. The e.m.f. registered by a multiple arrangement in which the temperatures at the junctions differ will be the algebraical mean provided the electrical resistance of all branches are the same. The proof is given in Appendix B, as it is perhaps not always recognized in connection with thermocouples.

Table 1 gives data on the plates made to date.

Since the manufacture of the plates required a large number of thermo-junctions it was necessary to have an easy way to make them. A little device shown in Fig. 6 was very successful and being semi-automatic relieved the nerve and eye strain. First the copper and then the constantan wire are pushed through the slot A , coming up against the stop B , and are clamped by turning the pin C . Stop B is then swung out of the way and the handle D given a complete turn, which brings

up the carbon and the spark makes the bead. Pressing lever *E* permits of the wires being removed. It has the advantage that it makes small beads and does not burn the wires. The wires are cut to length in a razor blade special device, are welded in pairs, and then the pairs in strings to form one row. All beads are squeezed in a small pair of pliers so that they are near the diameter of the wire in thickness and will sit flat on the plate, and while held in the pliers the constantan is bent so that a straight wire is formed, see Fig. 7.

The holes in the plates are drilled from a template with No. 56 drill. A needle at each end of the strings is used to thread them through, when they are shaped to

TABLE 1. DATA ON PLATES TESTED

Plate no.	Material	Thick- ness approx. inches	Differential Couples			Surface Couples ^a		Winding diagram figure	Finished weight lb.
			No.	Wire B&S	Resistance ohms	No.	Wire B&S		
1	Cork flooring	1/2	64	35	258 series	5	35	4	5.8
2	Cork flooring	1/2	64	35	256 series	5	35	4	5.8
3	Ebony wood	1/2	64	35	257 series	4	35	4	...
4	Ebony wood	1/4	100	35	510 series	5	35	4	1.6
5	Celeron	1/4	100	40	1060 series	5	40 & 35	4	8.1
6	Formica	1/8	196	40	1420 series 355-2P	5	40 & 35	4	4.2
7	Formica	1/8	196	40	1456 series 364-2P	5	40 & 35	4	4.5
8	Ebony wood	1/8	196	40	1428 series 357-2P	5	40 & 35	4	5.5
10	Cork flooring	1/8	196	35	16-4P	5	35	5	5.8
11	Formica	1/8	196	35	199 series 50-2P	4	35	5	4.8
12	Formica	1/8	196	35	161 series 40-2P	5	35	5	5.0
13	Formica	1/16	196	35	164 series 41-2P	4	35	5	2.9
14	Formica	1/16	196	35	160 series 40-2P	4	35	5	2.9
15	Formica	1/8	98	35	164 series 41-2P	5	35 & 28	5	4.6

^a The surface couples are in series; except boards 13 and 14, which are in parallel.

their correct positions. The ends of the strings are spark welded, the wires cemented to the plate with thin shellac, and the holes filled up with a mixture of shellac and chalk. The surface couples were then put on and the plate given a coat of spar varnish. The leads were added and three more coats of varnish applied.

If the plates could be left in this state it would be an advantage as any breaks in the wires could be repaired. There is not, however, a sufficient covering over the wires to reduce to a minimum the possibility of the temperature of the junctions relative to that of the rest of the surface varying with the method of heat application. To the cork boards was added a 1/16 in. layer of Acco cork, and to the others a layer of 4 lb. long-fiber asbestos paper, both being again varnished. The cementing on without special presses and steam plates was difficult to do well, but Armstrong Cork Co. medium cement, having a shellac base, was used for most of them and bakelite varnish for others. The latter showed signs of attacking some of the varnish coats, and it is probable that the more satisfactory way would be to use this after the wires have been lightly cemented in place.

Troubles due to lack of experience and care were naturally rather numerous while making the earlier plates, but even when using the No. 40 wire they can be made with a very small chance of failure later. Two of those made have failed, but it was in both boards due to easily preventable causes.

Calibration of Plates—Apparatus. The only method available for calibrating the plates is to pass known amounts of heat through them and find the e.m.f. registered by the couple system. Since these readings will be different for different plate temperatures, it is necessary to have flows at different temperatures. It was also evident that the work of calibration would involve the accuracy of the testing method as well as that of the plates, and that, as an investigation, the work on this feature was as important as the plates themselves.

The guarded hot plate transmits its heat from both sides, and though the total is known its division is not. Assuming that the plates themselves will always give the same reading for the same flow at a given temperature, it is possible to divide the plates between the two sides, and by interchange and adjustment to get two

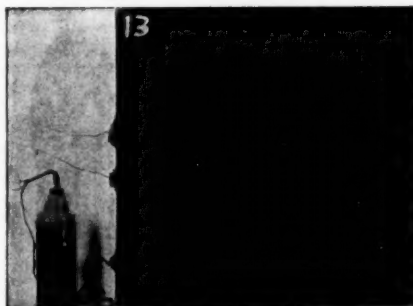


FIG. 8.

sets of values from which the ratio of the heat division could be determined. Single flow hot plates have been used but always on the principle of heavily insulating one side with a material whose conductivity was supposed to be known. This was not considered sufficiently accurate, so a new and simpler method was planned and has proved most satisfactory.

The method consists in using two ring guarded hot plates with a meter plate between them as shown in Fig. 9. The main hot plate *A* is the one to which there would be a constant supply of heat; the auxiliary hot plate *B* would be supplied to such an amount that the e.m.f. of the differential couples of *C* would be zero, that is there would be no flow of heat through the balancing plate *C*, and all the heat supplied to *A* would flow in one direction and that of *B* in the other. The plates to be tested would be placed on the *A* side, and on the *B* a block of material to limit the flow from *B*. There is no reason why the *B* side should not be used for testing except that it would require a longer time to reach a total equilibrium as the heat supplied to *B* has to be varied until that time.

The balance plate *C* was plate No. 10 of Table 1, made from $\frac{1}{2}$ in. cork flooring, with 196 differential couples. There were divided on a center line into two sections, and each half was again divided and made into two circuits in multiple. The

usual method of balancing was to have the two sections in multiple. A zero reading would indicate that on the average there would be no flow between *A* and *B*. If readings for the two sections separately showed that there was a flow from *A* to *B* in one half, then an equal from *B* to *A* would be given by the other. In all the tests, when constancy was reached, the largest departure of the halves from the mean zero was equivalent to 0.18 per cent of the rate of flow through the plate, based on per square foot values. The smallest was 0.02 per cent, and the average of all tests 0.1 per cent. The unbalance tended to be proportional to the rate of heat flow being used. The largest average difference in temperature between the two sides of one half of the balance plate was 0.04 deg. fahr., and the average in all tests 0.015 deg. fahr. Since the counter flows in the two halves balance each other, normally, they should introduce no error, but it is conceivable that they might, though the amount should be decidedly less than the above percentages.

These divided circuits were also some check as to whether there were any extraneous e.m.f.'s in the lines. Normally the series reading of the two circuits

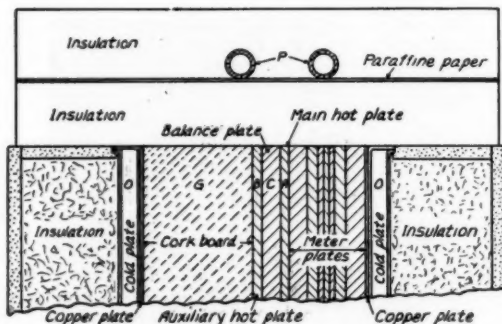


FIG. 9.

should equal twice the multiple reading, which should equal half the algebraic sum of the individual readings. This law was always found to be followed rigidly except in some tests when there was trouble in the instrument circuit due to moist air conditions.

As a final test of the accuracy of the balancing, a set of plates which had been calibrated by a one side flow could be divided into two sets and placed each side of one hot plate when the total B.t.u. flow obtained indicated on the two sides should equal the total input to the hot plate. Results obtained will be given later.

A further improvement could be made in the balance plate by putting differential couples around the guard ring part, dividing these into four sections corresponding to the four sides and testing for this further balance, and for further analysis the main winding could be subdivided into more sections.

The reliability of such a plate is shown by this one having been used at 230 deg. fahr. and up and down the scale without any trouble occurring.

Incidentally there is no reason why the same method should not be used for high temperature refractory testing, for which the cork board would be replaced by a block of pressed calcined diatomaceous earth.

The hot plates were made as per Fig. 10. The ingenious winding scheme of the Bureau of Standards was used as it gives the most even distribution of the heating wires. Advance strip was employed and the sizes fixed so as to give the most economical resistance for the voltages available. While this form was adopted it was not felt that it was ideal in indicating freedom from heat flow between the center and guard ring, and it is believed that such discrepancies as were found in the tests were due largely to errors in this balance. At the best the e.m.f. registered by the copper plates and a constantan wire joining them is one couple, and thus putting a large number of constantan wires only better averages without increasing the reading. This one couple will then be largely influenced by any stray e.m.f. which, considering the nearness of the high voltage windings, is quite a possibility on occasion. Moreover, though a zero reading for the copper plate may indicate no flow between them, yet there may be flow along the windings themselves or along the sheets of insulation. Another weakness is that it does not permit of balancing the four sides separately.

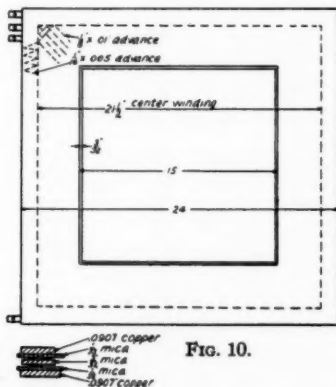


FIG. 10.

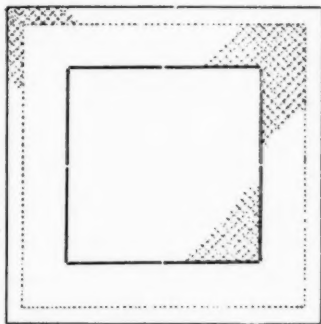


FIG. 11.

Fig. 11 shows a diagram of a contemplated form of plate which may be an improvement. The center part is separate from the ring except for narrow connections in the copper plates to hold the parts together. There would be three windings, the center one having leads to measure the voltage across it, a second one using the same size strip as the center and with the same distribution per unit area which would be in series with the center, and a balancing winding around the edge in which the input could be varied for each side.

In the slots, which are now open through the plate, would be strips of mica or formica which individually would be wound differentially with a number of thermocouples in series, and would thus balance the difference in temperature for the whole thickness of the plate, and would also allow balancing each edge separately.

That, in spite of the greatest care in making the hot plates uniform, balancing is difficult was shown in some of the earlier tests. Constantan wire was used on one side only and this was turned away from the plates under test. After several tests the hot plate was turned the other way and in all new tests the plate indications were two per cent lower than the former. After this balance wires were

used on both sides. It is possible that the relative unbalancing of the two sides is greater with two hot plates, due to the falling temperature gradient along the section of the plates starting from one side instead of the center.

The design of the cold plates is shown in Fig. 12. It consists of a main plate of copper with edges bent up to form a dish with a cover plate fitting inside. The guides to get an even distribution of water over the area was a continuous strip of $\frac{3}{4} \times \frac{7}{16}$ in. rubber. Small screws to draw the plates together ran through the rubber, thus making a tight joint both between the plates and at the holes. The only soldering necessary was around the edges to join the two plates, which could be done without warping them. The vents to prevent air pockets are shown.

The whole scheme of the assembly of plates under test is clear from Fig. 9. The backs of the cold plates were well insulated as were the outside edges of the whole

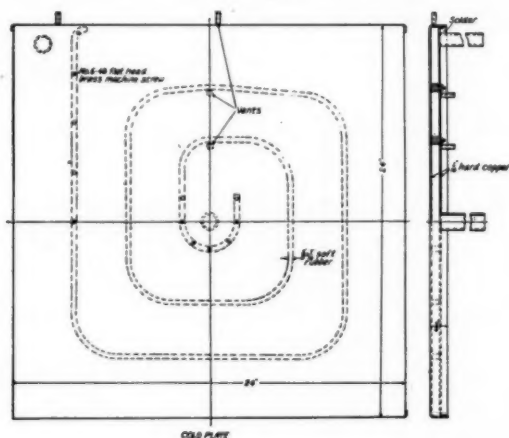


FIG. 12.

set up. The pipes *P* represent rubber tubing through which it was found necessary to circulate the cooling water, after it had passed through the cold plates, in tests where the interior material, including the hot plates, were much below the air temperature, since without these the heat inflow from the air was great enough to prevent a balance between the copper plates, even with no current in the outer windings.

Fig. 13 shows the scheme of control. All the resistances were of the tube type and each had in series with it a slide wire for making final adjustments. The ammeter and voltmeter were used for approximate readings, and exact readings were taken on the potentiometer, except for the amperes in the balance rings which were not required.

The potentiometer was a Tinsley Vernier reading direct to one microvolt. This was connected to the numerous lines through two pole mercury cups, which gave a very flexible arrangement for reversing the voltage and for the multiple and series changes which were needed for the boards.

No automatic regulation was attempted and a test when started would be run day and night with an attendant regulating it all the time.

The method of operation was to get to the approximate temperature using the power circuit, keeping the balance adjustments approximate, and then throw over to the battery. From that time the current through the center winding of the main hot plate would be kept at the value fixed for that test, measuring it on the potentiometer, and all adjustments made on the other circuits. As the battery capacity was limited the regulation of the main center current needed most attention, and, depending on the constancy of the voltage, would be adjusted every fifteen minutes or more. This was done by adjusting resistance *A* of Fig. 13 which, as it controls all the other resistances, does not change the balance. The current normally would not be permitted to change more than 0.05 per cent, and then, when ad-

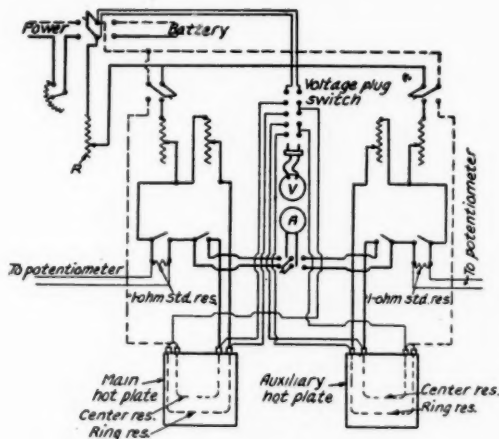


FIG. 13.

justed, it would be brought as much higher as it was found low so that the total input error would usually tend towards zero. The balance of the auxiliary plate temperature was handled in the same manner.

The constancy of the plate guard rings could be detected to 0.5 microvolts but lack of sensitivity in the galvanometer prevented closer adjustment. A change of one microvolt could easily be recognized in the change in the meter plate readings.

In the majority of the tests absolute constancy was not attained due to the time needed. Some tests, however, did not show variations in the differential readings of more than one in 5000 over a period of four to six hours. In others the run-up test would cross the run-down without the cause being located. Constancy at the hot end was naturally more easy to reach than was that at the cold end.

Because this constancy was attained it does not follow that it represents the accuracy of the results, in fact for individual tests it certainly does not, but on the other hand, it should create more confidence in the average of the results.

A test for one input took from 20 to 100 hours. When there was no guide as to the probable values it was necessary to run up towards constancy, then heat up safely above it, run down and take the mean. After the first set of plates had been calibrated it could be done more quickly as their values were a guide.

Fig. 14 is a diagram of the cooling water system. The main tank *A* is 24 in. x 24 in. x 30 in. inside and has a motor driven stirrer *B*. The pump *C* draws water from it, circulates it through the cold plates and returns it to the tank. The thermo-junction *D* has five couples in series and measures the temperature of the water as it leaves. When ice water is used, the stirrer is removed and broken ice about 18 in. deep is kept in *A*, maintained well packed down and not floating; there was no difficulty in keeping the exit temperature constant. When warm water was used its temperature was fixed at something above room temperature, so that the loss of heat from the tank would, preferably, be a little more than that added by the heat from the plate system. Ice water from tank *E* could be added.

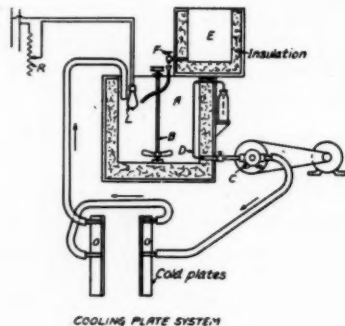


FIG. 14.

through the fine adjusting valve *F* which had a slight flow glass, or heat added by lamp *L*, controlled by resistance *R* placed on the main switch board. The regulation with warm water needed more attention and was operated on the swing plan. It could be kept to plus and minus 0.05 deg. fahr. The pump passed 18 lb. of water a minute, but an increase in this would be of advantage with the larger inputs.

Fig. 15 shows a photograph of the apparatus. Since constant attention was needed it was necessary to make the work easy. All adjustments, except that of attending to the water circulation, can be made without the observer moving from his seat.

Results with the Meter Plates

To discover what was the true reading given by meter plates for a given rate of heat flow, it was necessary to be able to separate errors in the estimation of the flow from those of the plates themselves. On the other hand the readings given by the plates were the only check available on the relationship of heat flow to total input, so that assuming the flow indicated may be either above or below the true value, it was necessary to make a large number of tests and to judge by analysis as to what errors were caused by plate, apparatus or operation.

By an error in a plate indication is meant that it can, for the same rate of heat flow, give more than one differential reading for each surface reading. Such might happen from the following causes:

1. The relative distribution of flow over the junction points and plate area differ. That this will be so to some extent is certain and can only be reduced by increasing the number of couples.

2. The plate has a different apparent thermal conductivity due to physical change in it, temporary or permanent. This involves repeated tests, and finding

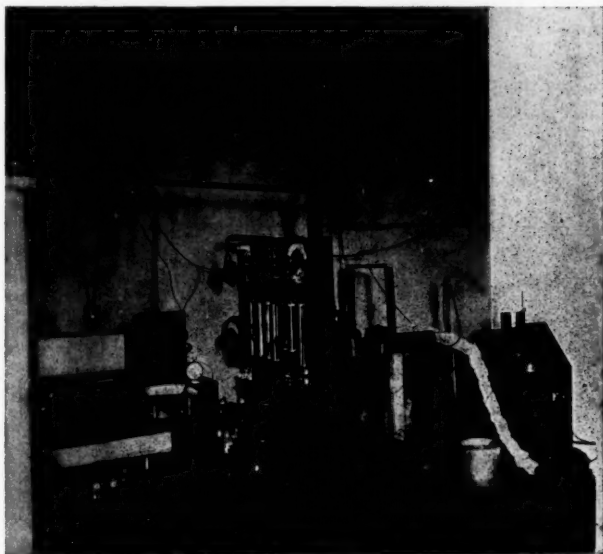


FIG. 15.

a material and construction that do not change with time, and the aging at a sufficiently high temperature to give it the final form.

3. The e.m.f. of the couples differ due to strains or leakage. The probability of this being anything appreciable at the temperatures for which they are intended, is not great.

In the first set of tests plates 1, 4, 6, 8, 7, 5 and 2 were arranged in that order starting from the hot plate. They had thin waxed paper between them to reduce the chance of sticking together but otherwise had close contact, and the edges were covered with tape, putty and paint to prevent air leaking between them. Such action would be shown by an apparent loss of heat, that is, lower readings of the meter plates, as the cold plate is approached.

It will be seen from Table 1 that the boards tested included two of $\frac{1}{2}$ in. thick cork flooring, specially selected and machined to size, supplied by the Armstrong Cork Co., one $\frac{1}{4}$ in. and one $\frac{1}{8}$ in. thick Ebony wood, a cement-base

asphalt-impregnated material, supplied by the Johns-Manville Co., two $\frac{1}{8}$ in. thick Formica, a Redmanol paper product, supplied by the Formica Insulation Co., and one $\frac{1}{4}$ in. Celeron, a vulcanized fiber, supplied by the Diamond State Fibre Co. In comparing results from such an assorted set of boards it would be expected that if all the boards showed the same general lack of concordance, then the cause would be due to the heat transmission through them, actual or deduced,

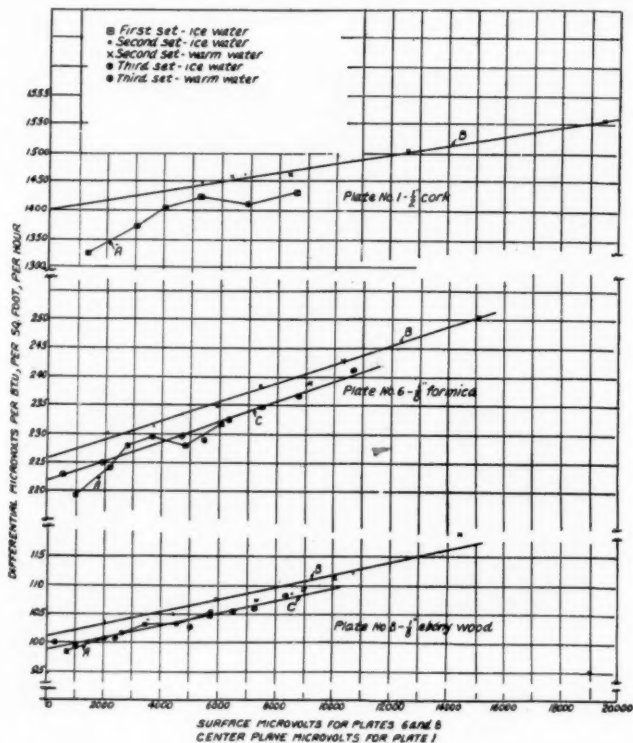


FIG. 16.

not being what it was supposed to be, and the source of error should be in the testing method or apparatus. If it were found that the discrepancies were between the different boards, some agreeing with the average and some not, then this would be due to the boards registering incorrectly. If results from any one test showed a probably correct value for one end of the assembly, and a gradually increasing departure from correctness towards the other end, then the most reasonable explanation would be that the heat was not following a straight line but was diverging or converging relative to the axis of the set up.

A complete set of results from three plates 1, 6 and 8 will be used as illustrative.

The method of plotting is to divide the differential millivolts by the B.t.u. transmission per square foot, and to plot this against the surface millivolts. This is equivalent to plotting temperature drop per B.t.u. against material temperature. The relationship will be discussed later.

In Fig. 16 the results of three sets of tests are given. By a set is meant tests in which there was no material change in the arrangement. The points connected by the irregular line *A* had balancing constantan wires between the center and ring copper of the main hot plate on the side away from the meter plates. In those averaged by line *B* the main hot plate was reversed and the balance couple was next to the hot plate, and in *C* (No. 1 plate was not included in this) there were couples on both sides. Corresponding test points can be found by counting from the vertical axis, and the similar deviation of the points will be noted, indicating that the deviation is caused by test troubles and not the boards. This is particularly clear when the relative position of the average lines are considered, as they show that the cause of the deviation is common to all the plates. Also the displacement of line *C* from *B* is the same percentage of the ordinate for both plates 6 and 8. The tests from which curve *A* is plotted were the first made, when the technique of best operation had not been acquired. It would be logical that they be between curves *B* and *C*.

The plates were now divided and placed each side of a single hot plate, from which the heat flowed in both directions, plates 6 and 8 being on opposite sides. The sum of the flows given by the readings of the two plates should be equal to the total input.

Three tests were made and the results for plates 6 and 8 are summarized below. The B.t.u. for the plates are from curve *C* of Fig. 16. As the heat flow given by the plates is the larger it indicates that the true curves for plates 6 and 8 should be one half per cent higher than those of curve *C* in Fig. 16, which is in line with logical expectations.

B.t.u. to hot plate <i>a</i>	Diff.	Plate 6		Plate 8			Comparison		<i>e</i> %
		Surface	B.t.u. <i>b</i>	Diff.	Surface	B.t.u. <i>c</i>	<i>b</i> + <i>c</i> <i>d</i>	<i>d</i> - <i>a</i> <i>e</i>	
22.85	2720	7294	11.6	1186	7537	11.14	22.74	0.11	0.47
47.63	5734	9170	24.12	2525	9694	23.3	47.42	0.21	0.44
72.96	8865	11377	36.75	3942	12119	35.7	72.45	0.51	0.70

This line of argument rather befogs the deductions as to the accuracy of the balancing plate in giving a one-sided flow. As they have been made on the assumption of it being accurate, as the conclusions are logical the assumption is justified. Apart from such assumption its accuracy has been proved to within one half of one per cent.

Fig. 17 shows the curves for all plates in this first investigation arranged, from the top, in their order relative to the hot plate. The average lines have been drawn in by eye. It will be noted that the agreement of the points with the mean decreases as the cold plate is approached, and that there is then a tendency of the tests with ice water in the cold plate to cross those using warm water. These actions could be explained by a lack of temperature uniformity in the cold plate, or more probably as due to the thermal resistance of the built-up mass of plates not being uniform.

One differential circuit of plate 7 failed during the tests leaving only 100 couples, otherwise this would have been more uniform.

Plate 2 is similar to 1. Its values with the ice cold plate may be influenced by moisture since it was at a low temperature for several months.

The next set of tests covered five new plates, numbers 11 to 15, with which plates 6 and 8 were included as a guide in testing. This made the work easier and the complete tests were made in under two weeks. The results are shown in Fig. 18. Some of these tests were made under difficulties that did not permit of them being carried on as long as they should have been, and this is clearly shown in the common departure from the average of corresponding points, and thus is a test error.

There is an indication that plates 6 and 8 give more consistent results than do the others, which would be attributed to the number 40 wire used for the couples.

This set of plates was also tested by dividing them into two sets and using a single hot plate, giving the following results, all heat flows being per square foot per hour.

FIRST SIDE			SECOND SIDE		
Plate	B.t.u.	Departure from mean %	Plate	B.t.u.	Departure from mean %
13	22.30	-0.76	11	25.59	+0.8
14	22.47	-0.04	6	25.46	+0.3
8	22.52	+0.18	12	25.1	-1.1
15	22.62	+0.62
Mean	22.48	...	Mean	25.38	...

Total input from plate readings, 47.86 B.t.u.

Total input actual 47.672.

Difference 0.42 per cent.

The percentages can be taken as a probable fair measure of the accuracy of the plates and an indication of the lack of absolutely uniform distribution of the heat flow in different tests.

Since the readings from the plates both in test and application are in e.m.f. it is very desirable to use these values instead of converting them to temperatures, thus requiring a second conversion. Moreover by using the e.m.f. no assumption is made as to the relationship between it and temperature, or as to variations in this due to changes in the wire. The only assumption made is that a given couple will always give the same e.m.f. at the same temperatures. Since this is a new language it is well to find what theoretical relationship should exist between the differential value per B.t.u. and the surface e.m.f. This is covered in Appendix C, which shows that the relationship is very closely a straight line within the range of temperature used and supports the experimental curves.

The slope of the curves of Figs. 17 and 18 reduced to the same basis as that of the Appendix, that is, the curve slope divided by the ordinate reading at zero surface microvolts, and then multiplied by the number of couples in series, should give the same value for all plates, of the same material and using the same wire for the couples. The following tables give those that can be compared:

Plate	Material	Thickness in.	Wire no.	Slope
11	Formica	$\frac{1}{8}$	35	5.04×10^{-5}
12	Formica	$\frac{1}{8}$	35	4.9×10^{-5}
13	Formica	$\frac{1}{16}$	35	4.9×10^{-5}
14	Formica	$\frac{1}{16}$	35	4.85×10^{-5}
15	Formica	$\frac{1}{8}$	35	3.5×10^{-5}
1	Cork	$\frac{1}{8}$	35	2.1×10^{-5}
4	Ebony wood	$\frac{1}{4}$	35	5.6×10^{-5}

The first four show close agreement, but board 15 is out. This had been overheated during its making and not much reliance can be placed on it in such a comparison.

The constantan wire was of a normal grade and gave in calibration an equation of $e = (21.58)T + (0.013)T^2$.

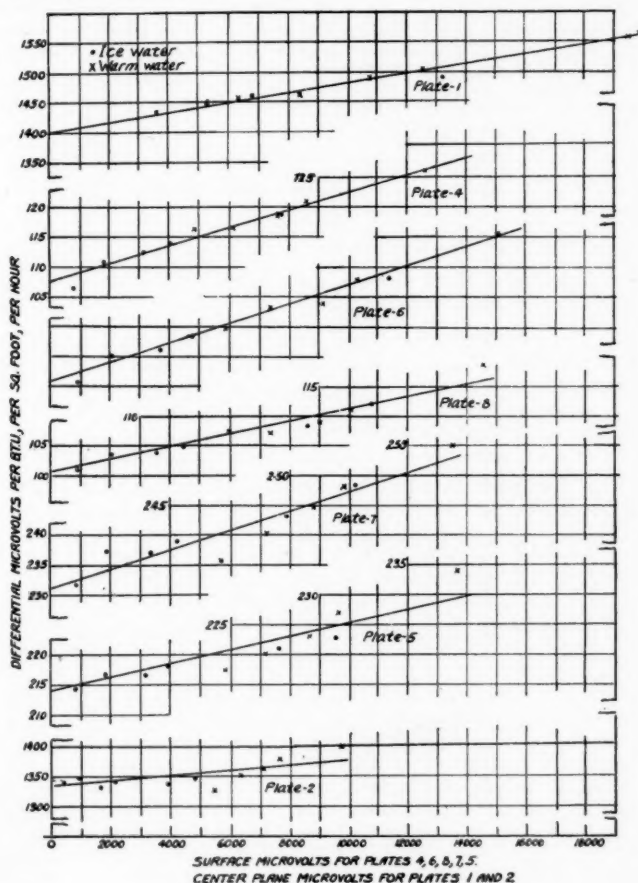


FIG. 17.

From this and equation 7 of Appendix C the following conductivity equations for the three materials can be obtained.

$$\text{Formica, } k = 1.63(1 + 0.0000637T)$$

$$\text{Ebony wood, } k = 3.21(1 - 0.000087T)$$

$$\text{Cork flooring, } k = 0.496(1 + 0.000687T)$$

T being in degrees fahr., and k in B.t.u., sq. ft., in., hr., deg. fahr. Not too much reliance should be placed on the first figures due to the thinness of the boards, and the relative importance of the size of wire. The multipliers for T are of the relative order that would be expected.

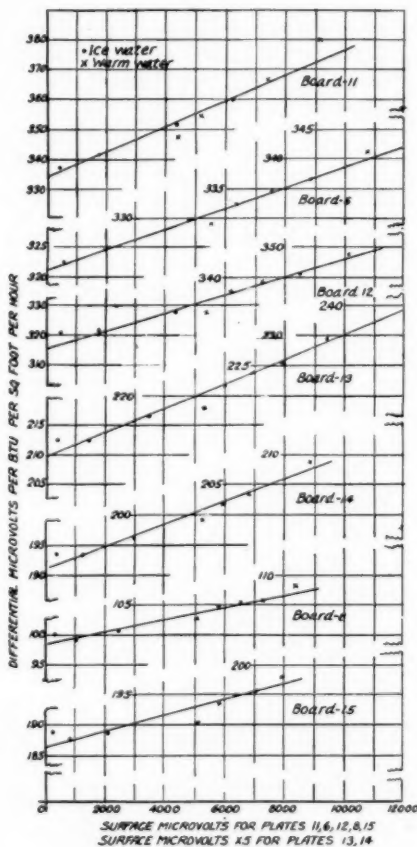


FIG. 18.

In the tests up to this stage the boards had been assembled touching each other. However small the wires may be they will tend to upset the uniformity of heat flow, and the relative amount of disturbance may differ depending on the way the heat reaches the surface. With the small wires used it was believed that the covering placed over them would be sufficient to distribute it so that the same proportions would always flow through the couple points. To test this the second

set of boards were assembled with an air gap of 0.11 in. between them by cementing strips $\frac{1}{4}$ in. wide around the edges. When assembled the edges were painted with melted paraffin to prevent air circulation.

These tests have not been carried far, and doubt was felt at making them at all without being able to balance the individual sides of the edge guard rings of the hot plates. Two tests were made with 18 B.t.u., sq. ft., hour flow, one using ice water and the other water at 82 deg. fahr. The summarized results are given in the table and compared with those in the previous tests.

Plate	WITH ICE WATER			WITH WARM WATER			
	Microvolts per B.t.u.		Per cent difference	Microvolts per B.t.u.		Per cent difference	Mean per cent
	With air gap	Without		With air gap	Without		
11	348	355	-2.0	372	369	-0.8	-1.4
6	230	230	-1.6	239.7	236.5	-1.3	-1.4
12	322	330	-2.3	345.5	337	-2.5	-2.4
13	203.5	217	-6.4	277.8	214	-6.1	-6.3
14	183.7	196.5	-6.5	206.2	193	-6.4	-6.4
8	98.3	101	-2.7	106.2	102.8	-3.2	-2.9
15	187.7	188	-0.2	195.3	195.4	-0.5	-0.4

In these tests all the plates had air gaps, so that no one value is obtained under the former close contact. They are given to show that it is possible that the method of manufacture has not been accurate enough to eliminate variations in thermal resistance due to the wires. If this is confirmed it will be necessary to apply a multiplying factor for use when one face is exposed to the air, as it is not desirable to increase the total resistance.

The absolute accuracy of all values depends on the error involved in translating the electrical input into B.t.u., and the measuring of the former. The total length of strip used on the center winding of the hot plate was known to 0.05 per cent, and its distribution on the surface can be calculated from the system of winding. Thus the ratio of the length per square foot area to the total length is known. Measuring the voltage drop across the total length in each test, with the fixed amperes flowing gives a means of obtaining the watts per square foot of plate area. The voltage was measured on the potentiometer using a megohm and post office box in series, the latter to obtain suitable values for the potentiometer, and Fig. 19 gives the apparent resistance of the center winding against the approximate temperature of the hot plate as obtained from the various tests. This shows a decrease of apparent resistance with rise in temperature and gives a more consistent result than does a plotting against the input. The shunt resistance for measuring the current was checked against a recently certified standard ohm. From all checks the error in the conversion to B.t.u. is not more than 0.1 per cent, this of course not including the disturbing of conditions by edge-wise flow, nor does it bear on the distribution of the flow through the material tested.

The order of the error due to using the mean e.m.f. readings of the couples instead of the mean temperature is worked out in Appendix B and is found to be negligible even for extreme unbalance.

Comparing the results from the thick and thin plates there is no indication that the former are more consistent or reliable. Such comparison is not entirely justifiable in that the thinner boards had more differential couples than the thick.

Nor is there much in favor of any one material. Cork flooring was used because it gives a high reading and has the small heat capacity of 0.93 B.t.u. per deg. fahr.

per sq. ft. per in. Such a material would be ideal in its properties but its flexibility in thin sheets would be dangerous for the small wires, and any material which has air spaces must necessarily be more liable to change with time.

Ebony wood has a low thermal resistance, which causes a smaller differential reading for a given thickness. On the other hand it is heavy and has a higher heat capacity of about 2.6 B.t.u. It is very rigid even in thin sheets and this may

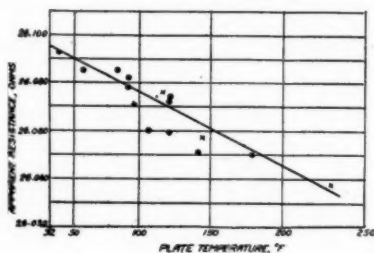


FIG. 19.

or may not be an advantage. From the nature of its manufacture it would be expected to lack—in thin sheets—that exact uniformity possessed by products made from one material and without grain, but the two samples tested give very satisfactory results.

Formica, or similar products, have a higher thermal resistance, are strong but flexible, and the uniformity in thermal properties should be good. Its heat capacity is about 2.4 B.t.u. Below $\frac{1}{8}$ in. thickness it tends to have some buckle and no change with time was noted, but a similar material and a little stiffer would be an advantage.

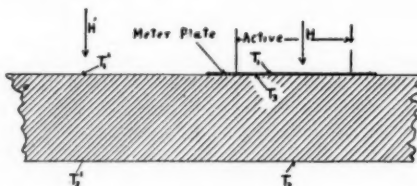


FIG. 20.

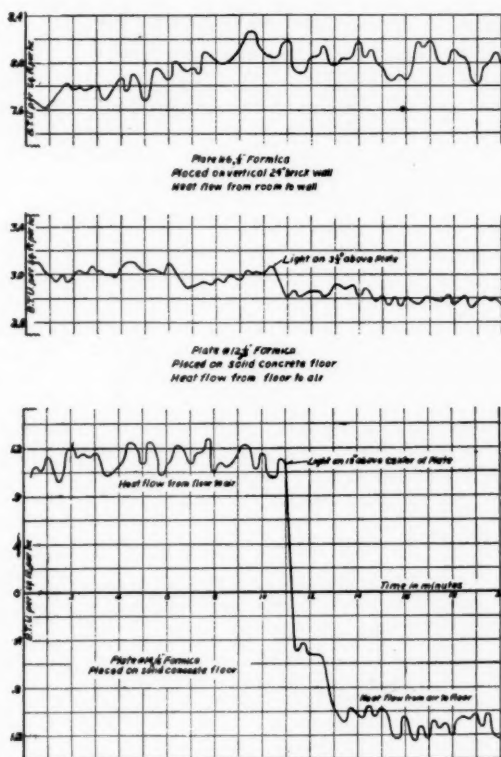
As a brief summary it can be stated that:

1. The plates as constructed after a nine month's use have shown no tendency to change their reading for a given flow due to change in the plates themselves.
2. That with close contact on two sides the readings they give can be taken as a true measure of the heat flow to within plus or minus one per cent.
3. That to obtain closer accuracy it will be necessary to attempt to eliminate some doubtful features of the method of test, and to institute further checks.

Plates as a Meter of Heat Flow. A plate will at all times give the heat flow through the active portion, that is the area covered by the differential couples, and will

average this if the flow is not uniform over the area. It will however add a thermal resistance to that area, and thus will make the flow less than that through the natural wall.

In addition to adding this resistance it also tends to change the parallel flow through the wall, and in order that the flow through the active part be a true



FIGS. 21, 22 AND 23.

measure of the heat flowing through the wall covered, the guard ring part of the plate should be wide enough to keep the flow in parallel lines. The actual guard ring of the plates is $4\frac{1}{2}$ in. wide, and where exact data is desired it will be necessary to increase this by adding strips of equal thermal resistance.

An exact treatment of the relationship of the flow indicated by the plate, considered as an approximately instantaneous value, to that through the uncovered wall, is complicated and for practical work of no interest where the flow is variable. Assuming the wall is uniform the flow, H , Fig. 20, through the uncovered portion

is, $H^1 = H \frac{T_1 - T_2}{T_3 - T_2}$. If all the conditions are constant then one set of readings would give the data, but as this is rarely so, the H and the temperature would have to be the sum or mean of the values over a sufficient length of time, which as a minimum would be that between two similar temperature conditions of all the wall.

Similarly if the thermal conductivity of the portion of the wall covered by the plate is to be studied, its average value will be given by $k = H \frac{\text{thickness}}{T_3 - T_2}$, with values taken in a similar manner and averaged over the time.

Used to measure the fluctuating flow of a basement floor, tunnel wall or otherwise solid mass of material with constant or slowly varying air temperatures, the flow given by the plate will be sufficiently accurate with that of the natural body. If further checks be desired a portion of the surface would be painted the same as

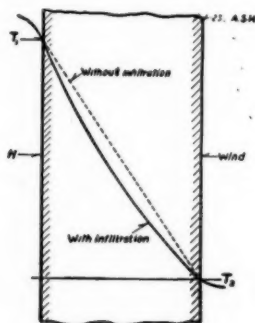


FIG. 24.

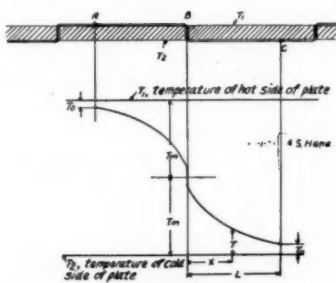


FIG. 25.

the board and temperatures of the painted and unpainted portions taken. In any application the plates must be in position long enough before readings are taken for the temperatures to have readjusted themselves.

The total thermal resistance added to a wall by a plate is dependent on the closeness of the contact, and the surface should be level and smooth. Tests of plates on a plaster finish which was not free from waves gave the following values:

Plate number.....	12	14
Nominal thickness.....	$\frac{1}{8}$	$\frac{1}{16}$
Thermal resistance surface of plate to surface of wall, R	0.305	0.216
Ratio of R to that measured by the differential couples.....	3.8	4.7
Equivalent thickness of brick, $k = 6$, having a resistance R	1.83"	1.3"

Application of plates to a variety of conditions in order to test the type of reading they give brought to light an action which is more accentuated than was expected. Supposing a steady flow of heat into a surface, then a very small instantaneous change in temperature at the surface will produce a very large instantaneous change in

the rate of flow, which will however quickly die down to a small normal change. As the plates measure the temperature difference through an appreciable thickness the initial change is masked, and a smaller change of flow is shown than occurs at the true surface. With a natural convection and the generally accredited stationary air film it was expected that there would have been a further lag, and that the wave length of changes would be large.

Curves are shown of readings at one quarter minute intervals. The base line is shown for one, but the side scales fix the range of the variations relative to the total flow. Fig. 21 is for a plate on a vertical 24 in. brick wall. Due to room conditions there would be more air motion and temperature variation than usually found. Figs. 22 and 23 are for plates on a concrete floor, solid with the foundation and with steady room conditions. It will be noted that the $\frac{1}{16}$ in. plate registers larger fluctuations than does the $\frac{1}{8}$ in.

To illustrate how rapidly the plates record new conditions, a 60 watt lamp was suspended above each of the floor plates. The immediate action of the radiation is very marked.

The plates are thus very sensitive in their record, but this same sensitivity increases the difficulty of taking readings, which have to be averaged. It would thus be of decided advantage to bury the plate a small distance in the wall.

Observations on any one trial application of the plates were not sufficiently continuous to deduce values of constants, but they clearly showed the change in flow with conditions and the lag due to the mass of the wall.

It is plainly desirable that the various readings should be automatically recorded. There are various instruments that could record the temperatures, but the comparatively high resistance of the couple circuits makes a potentiometer type very desirable. In many applications the only values required would be the sum of the readings and, if it could be devised, an instrument similar in principle to a voltmeter would be ideal.

In what way can such heat transmission meters be of assistance? Their primary use will be to give records of the flow which actually occurs, without regard to the cause, to obtain a check on calculated values, or to measure the loss of heat—temporary or average. Such records are needed both for the data they may give, and also to establish confidence in the ability to predict by calculations.

They can be used to measure conductivity coefficients of materials under natural conditions, but such work must include a full set of temperature measurements, including some through the wall if it is of compound construction. In as far as the temperatures are not constant such test will have to be continuous, or by automatic records. The need for and advantage of doing such work under natural weather conditions are that it is removed from the opprobrium of being a laboratory test.

They should be of decided value in connection with laboratory tests in that the heat input need not be measured. Any source of heat at a constant temperature can be used and the actual transmission can be measured at the plane or planes most suitable. In high temperature work the actual hot temperature, when it is generated electrically and the heat input measured, is limited by the ability of the apparatus to stand it, but by using a plate at the cold end the hot temperature can be produced by a flame. Using one each side of a single hot plate gives the division of heat on the two sides, which it is very desirable to know. They can also be a check on heat input, or of ice melted.

For natural wall transmission the plates are, in principle, inferior to Barker's box method, which disturbs the conditions less. They have the advantage however of greater simplicity and, it is believed, greater accuracy of measurement, especially if the room conditions are variable, as well as the ability to record flow in either direction. They become a necessity where observation must extend over a long period in order to cover a cycle of changes, such as the yearly one in a tunnel under a river.

Conditions Affecting the Apparent Thermal Conductivity

Assuming that the materials in a portion of a structure do not change, the thermal constants will also remain unchanged, but the heat transmission through it may vary from two causes imposed by and related to the weather conditions, namely the amount of moisture in the wall and on its surfaces, and also the exterior wind pressure in as far as it changes the surface transmission coefficient, or causes infiltration through the whole or a portion of the material. Both actions will affect the apparent thermal conductivity as deduced from the measured heat flow and will make it differ from values obtained by laboratory tests.

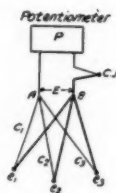


FIG. 26.

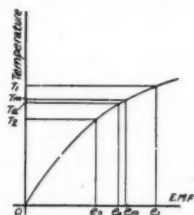


FIG. 27.

Moisture in a material will always increase its thermal conductivity, and that this increase may be very material is well recognized though there is little data on it. The National Physical Laboratory of England reports a test on a nine inch brick wall which when thoroughly wetted (the exact meaning of this not being stated) had a conductivity coefficient of 15.8, which, under the test heat finally reduced to 6.1. It is understood that considerable attention is being given to moisture in the Swedish test though no details have been published to date.

It is to be presumed that the moisture content of a wall will go through cycles, but that its amount and distribution will depend on a number of complicated factors. Its deposition will depend on the absorptive capacity and the driving force of rain; on the ordinary laws of condensation which will tend to collect moisture in the inner portion during the warmer weather, or at any portion after a rapid rise of air temperature; due to diffusion where the vapor pressures in and outside differ, and to a smaller extent due to temperature differential diffusion. On the other hand it will always follow the regular laws of drying by evaporation at its surfaces.

It can be expected that the rate of change, particularly the decrease, in the moisture content will be comparatively slow. When there is air infiltration it will tend to dry the wall in winter and usually add moisture in summer. Whatever moisture there is will be a part of the structure and the thermal conductivity measured at that time will be the true one.

It can be believed that the amount deposited by interior condensation will not be formed at a high enough rate to affect materially the thermal gradient by the latent heat given up, but on the other hand the evaporation at the outer surface will increase the surface transmission coefficient and abstract heat from the wall or keep the surface temperature down.

In a well-built thicker wall the amount of moisture content should be approximately the same during the same periods of different years, when it is averaged over a week.

Air infiltration due to the wind can, as far as its overall effect is concerned, be treated as a portion of the air change in the building, but it also comes into evidence when measuring the heat transmission of a portion of the wall.

The infiltration concerned with here is that air which leaks through due to the porosity of brick, concrete, and probably some stones, and that due to mortar joints which are evenly enough distributed to be considered uniform. Such infiltration evidently will affect the temperature gradient in a wall, and, depending on how the heat transmission is measured, will give an apparent conductivity coefficient which will differ from values found without infiltration.

A brief mathematical development is given in Appendix D and is on the assumption of uniform infiltration. Such infiltration has not been measured separately from other leakages, but recent work in this laboratory on window leakage obtained values for a brick wall with an area of 44.6 sq. ft., which is sufficiently large to be a fair indication of what such values may be. The wall was 13 in. thick with $\frac{1}{2}$ in. plaster, and one of the values will be used for illustration.

With a difference in pressure equivalent to a wind velocity of 29 miles an hour there was a leakage of 20 cu. ft. per sq. ft. per hr. Assuming that the temperature of the wall inside was T_1 , and outside T_2 , and that the conductivity of brick is $k = 6$, the estimated flow, if there were no leakage, is $0.445 (T_1 - T_2)$. With the infiltration the estimated flow measured on the room side is $0.66 (T_1 - T_2)$, and at the outside $0.28 (T_1 - T_2)$. The apparent conductivity coefficients would be 8.9 and 3.8, respectively. Fig. 24 gives the temperature curves through the wall with and without the infiltration.

The illustration is probably extreme for continuous conditions and is given to show what the order of values may be, and how natural conditions may give coefficients differing from those obtained in the laboratory. It also illustrates the difficulty in measuring heat flow without changing the natural conditions.

Review of Investigational Trend

It is pertinent to the intent of this paper to review briefly the present standing of our knowledge of and the data available for heat transmission in connection with heating and ventilating engineering. If there were agreement on what is lacking, it would fix the trend of future investigations. It is well to start with the statement, to which there probably will be no objection raised, that the investigational knowledge should be of a higher degree of accuracy than will be required for the majority of its applications, or of the ability to define conditions and should be in advance of practical requirement.

It is also taken for granted that the ultimate aim of the Society should be to issue under its approval a compendium of values and methods of estimation; that in this as in all other branches of technical calculation there will be the tendency for closer analysis and the more accurate prediction.

The truth and sufficiency of data are usually demonstrated by their application to practical problems being confirmed by the results obtained. In the heat losses here considered, not only has there been no means of checking as to whether those occurring agreed with those estimated, but they are only a part of the total heat exchange, and there is no means of determining the proportion they actually have to the total exchange as measured by the heat input. It therefore has been natural that there should be a lack of confidence in the correctness of the method of estimating or of the constants used. Such doubt is usually explained as a disbelief in laboratory tests on small samples representing the actual conditions in large structures. Such a statement is justified in as far as tests have not shown how values obtained under what may be called "stationary conditions" will vary with superimposed conditions that will occur in the structure. These conditions are moisture, rain and wind. Temperature is omitted, since the heat flow can be predicted against its cycles, at least to a fair accuracy.

It is this feeling of the lack of connection of laboratory tests with practice that has of late years encouraged the attempt to determine values of constants under natural conditions. The difficulties involved in this have been brought out in this paper, and it is probable that in the main they will give overall actual values without being able to connect them closely with the causes, because these do not maintain a given value for a long enough time.

From an investigational standpoint it would be desirable first to connect the distribution of moisture in a wall with the weather conditions, and also to determine the thermal conductivity of the material with each moisture content within the practical range. Such an undertaking would be large and impractical in as far as the material itself cannot be confined to a limited number of definite physical specifications. It would seem, however, that they should be carried on far enough to show the relative order of the values for a limited number of changes in order to prove or disprove the importance of the moisture feature. A knowledge of this is necessary to fix values for average use, and also to show the economy of waterproofing or other means of prevention.

The same general argument applies to wind pressure, but it is probable that the possibility and economy of prevention are the more important aims.

The need for such trend in the investigations are of more importance when tests are conducted to compare types of construction, as it is quite probable that one which showed the better under stationary might be decidedly inferior when subjected to natural conditions.

It is thus the author's belief that further tests to determine conductivity values for structural materials will be incomplete unless they are extended to include their behavior under either natural or imitations of natural conditions. In as far as this is not done scepticism will arise in the future as to their reliability as practical values.

There are two other factors which are probably largely the cause of discrepancies obtained by different investigators, the first being the wide range of physical qualities which may occur in a material which is classed under one nomenclature and considered as having approximately fixed qualities, the second that of the grade of workmanship in the assembly of the units in the structure. There has been little attempt to determine the relationship of the thermal conductivities of given structural materials with its other physical variables. This has been done for some insulating materials, but for these it is a simpler problem, since usually the structure is the same and has a definite form for each specific weight. With many building

materials this is not so, and an additional physical quality in the closeness of the cementing of the particles together comes in, such as in the burning of a clay product. Also the number, size, and porosity of the individual particles will materially affect its thermal conductivity. The materials coming under this classification are not numerous and it would be a praiseworthy undertaking to classify and determine their conductivities under stationary conditions, and certainly would be a more valuable addition to our knowledge than would a similar number of tests on diverse materials.

The investigational requirements would not end with the stationary tests, as these same materials might have different qualities of absorption and air infiltration. It is however quite a possibility that the change in thermal conductivity could be fairly accurately connected with absorptive qualities through deductions from results with one given material.

The grade of workmanship factor is of importance but is not so much an investigational problem. It will not usually affect stationary values materially but is of importance mainly in the quantity of infiltration. It may in some constructions influence the amount of moisture in the material, although both factors could be largely due to the architect's specification, such as the type and method of application of building paper, prevention of interior air currents, and finish of mortar joints. If there were any question as to the economy of a given grade of workmanship, the part that the relative heat transmission would play in such economy could be determined for a few typical constructions.

There is one further factor, namely the change in heat transmission with time. The largest part of this would be due to the deterioration of the structure as a whole rather than change in the specific conductivity of the materials. Even where the materials themselves are perishable, such as in frame buildings, the increased transmission due to defects in the structure itself will certainly outweigh any change in the material. In fact it would be expected that the conductivity of the dry material would be decreased, and an increase would only be due to its larger moisture content.

As the heat transmission of building materials is mainly of interest during winter weather, it would seem desirable that laboratory tests should be arranged to have one side around freezing point. As far as they try to imitate natural conditions, tests with a still lower temperature would be advisable at least for a limited number of comparisons, more especially if any attempt is made to duplicate natural moisture conditions.

Assuming that the investigations outlined were completed, it is granted that in the majority of practical computations an average value would be sufficient, and as nearly correct as the knowledge of other factors. It is to be presumed however, that the increase in knowledge of the other factors of window infiltration, health requirements, and ability to fortell the economic size of heating plant needed would have kept pace, and that a closer estimate on the heat transmission would be desired. Apart from this, one of the main advantages in such knowledge is in its influence on practice in showing how heat losses can be reduced and producing data from which the economic value of such saving can be deduced.

It is not germane to this paper to include a summary of recent tests, for quite extensive work has been carried out during the past five years in England, Sweden and Norway. A list of the original publications is given in Appendix E.

Acknowledgment is made on behalf of the laboratory to the various manufacturers who have willingly supplied any material requested. These include the

following companies: Armstrong Cork, Formica Insulation, Johns-Manville, Philip Carey, Keasby and Mattison, Rock-Products, and Driver-Harris Co. Particular thanks are due to E. C. Loyd of the Armstrong Cork Co.

The following members of the laboratory have in turn assisted in the experimental work: James Brown, E. C. Andrews, and Raymond Otter.

APPENDIX A

Relationship of Differential E.m.f.'s to True Surface Temperature

Fig. 25 shows a section of a plate of low thermal conductivity material through which heat is passing. Also a series of couples with alternate junctions on the two faces and passing through small holes. The wires are assumed to be cemented to the surfaces and covered with a layer of cement or material so that they are immersed in it. A and C are junctions.

Since the metal wires have a higher thermal conductivity than the material, heat will be conducted along them and through to the colder side more easily than through the plate, and thus the temperature of the wires will differ from that of the material surrounding it, and the differential temperature registered by the couples will be, in general, less than that of the surfaces.

The type of distribution of temperature is shown in the curve below the plate.

Since the two sides are symmetrical it is only necessary to consider one. Also since the plates are thin, the problem need not be complicated by taking into account the portion of the wire passing through the plate, or the length on the surface can be assumed to be increased by one-quarter the plate thickness. The temperature at B can therefore be taken as that of the middle of the plate = $\frac{T_1 + T_2}{2}$.

Considering at first only the portion $B - C$ of the wire, and assuming that the junction C is nearer to the surface temperature than any other point in it so that there is no heat flow along the wire at C , the portion $A - B$ is equivalent to a rod whose temperature is kept constant at B , and which is immersed in a medium of temperature T_1 .

Measuring temperatures T from T_2 , and distances, x , along the wire from B , the general equation for the temperature T at x is

$$T = A e^{cx} + B e^{-cx} \dots\dots\dots (1)$$

where A and B are constants to be determined by the conditions, and $c = \sqrt{\frac{4E}{Kd}}$

where

E = rate of heat exchange to medium per unit surface of wire per deg.

K = thermal conductivity coefficient of wire.

d = diameter of wire.

At the point B , $x = 0$, $T = T_2$

therefore $T_2 = A + B$

At the point C , $x = L$, and $\frac{dT}{dx} = 0$

But $\frac{dT}{dx} = A c e^{cx} - B c e^{-cx}$

From these A and B are determined as

$$A = \frac{T_m e^{-cL}}{e^{cL} + e^{-cL}}, \quad B = \frac{T_m e^{cL}}{e^{cL} + e^{-cL}}$$

thus

$$T = \frac{e^{cx} e^{-cL} + e^{-cx} e^{cL}}{e^{cL} + e^{-cL}} \times T_m$$

At the junction C , $x = L$ and therefore:

$$T_o = \frac{2T_m}{e^{cL} + e^{-cL}} \dots \dots \dots (2)$$

Considering the wire CD , which is of a different metal,

$$T_o = \frac{2T_m}{e^{c^1 L^1} + e^{-c^1 L^1}} \dots \dots \dots (3)$$

For (2) and (3) to be true, that is for the junction to be the point of the wire nearest to the temperature of the medium, cL must be equal to $c^1 L^1$, or

$$L \sqrt{\frac{4E}{Kd}} = L^1 \sqrt{\frac{4E^1}{K^1 d^1}}$$

The surface coefficients will be the same, and if $d = d^1$,

$$\frac{L}{L^1} = \sqrt{\frac{K}{K^1}}$$

The K 's for copper and constantan in foot-inch-hour-deg. fahr. units are 2650 and 157, therefore:

$$\frac{\text{length, copper}}{\text{length, constantan}} = 4.1$$

Taking the wire as a buried pipe, the zero surface temperature being at a diameter D , and conductivity of the medium k :

$$E = \frac{2k}{d \log \frac{D}{d}}, \text{ and } cL = \frac{L}{d} \sqrt{\frac{8}{\log \frac{D}{d}}} \sqrt{\frac{k}{K}} \dots \dots \dots (4)$$

$$\text{Writing equation (2) as: } \frac{T_o}{T_1 - T_2} = \frac{1}{e^{cL} + e^{-cL}} \dots \dots \dots (5)$$

The left hand side is the proportional loss of temperature due to the length of wire. Taking k for the medium as 2, and considering the copper half of the couple the following table gives values for (5) for assumed ratios of $\frac{D}{d}$.

Tests were made to check these values and determine the value of c , but as far as they were carried the results were not very consistent when using a heavy material board, due probably to the size of wire in keeping the junction away from the surface, being of more importance than the lengths used. Also with short lengths the stiffness prevented them being cemented as closely. They showed, however, that as large an e.m.f. could be obtained with 0.3 in. constantan as with longer lengths.

PROPORTIONAL LOSS IN E.M.F. WITH GIVEN WIRE LENGTH ON ONE SURFACE

$\frac{D}{d}$	Wire gauge B&S	Length of Copper Wire in Inches						
		No.	0.1	0.2	0.4	0.6	0.8	1.0
20	32	0.43	0.29	0.105	0.035	0.01	0.003	0.001
	35	0.37	0.17	0.04	0.008	0.0016	0.0003
	40	0.23	0.06	0.004	0.0002
50	32	0.45	0.32	0.135	0.05	0.018	0.007	0.0025
	35	0.4	0.23	0.06	0.008	0.0035	0.0009	0.0002
	40	0.26	0.08	0.007	0.0004

APPENDIX B

Thermocouples in Series and Parallel

Series couples in which every alternate junction is kept at a common temperature and the other junctions at various temperatures will give an e.m.f. which, divided by the number of pairs of junctions, will give the true mean e.m.f.

If the similar metals of a number of junctions be connected together and run to a common cold junction, the e.m.f. registered will be a true mean—for all temperatures—only if the electrical resistances of the parallel junctions are the same. The proof of this is as follows:

Let n junctions, Fig. 26, be joined at A and B and have a common cold junction CJ , which can be considered to be at zero e.m.f. If a potentiometer be used there will be no current in APB , and the voltage E across AB will be that registered.

Let the e.m.f.'s of the junctions be e_1, e_2, \dots, e_n . Since these are not necessarily equal there will be currents flowing in the junctions and through A and B . Let these be C_1, C_2, \dots, C_n , called positive if flowing to A and negative, if away. Let the electrical resistances be r_1, r_2, \dots, r_n .

Taking each junction in turn:

$$\begin{aligned} E &= e_1 - C_1 r_1 \\ &= e_2 - C_2 r_2 \\ &= e_n - C_n r_n \end{aligned}$$

$$\text{therefore} \quad nE = e_1 + e_2 + e_n - (C_1 r_1 + C_2 r_2 + C_n r_n) \dots \dots \dots (1)$$

If each resistance be equal to r , then

$$nE = e_1 + e_2 + e_n - r(C_1 + C_2 + \dots + C_n)$$

But $(C_1 + C_2 + C_n) = 0$, since all legs meeting at A are included,

$$\text{then will } E = \frac{e_1 + e_2 + e_n}{n}$$

that is, E is the true mean, whatever be the values of the e 's. It also can be shown that if a galvanometer or millivoltmeter be used, the current through the instrument will be that due to a true mean e.m.f. only when the resistance of the parallel junctions are equal.

It is hardly necessary to point out that in equation 1, the negative part could be zero with unequal values of the r 's, but only for definite relationships of the e 's.

Unequal resistances can introduce considerable error if the temperatures measured be different. For example, if there be two circuits having resistances in the ratio of 1 to 2, and with one e.m.f. twice that of the other, then the e.m.f. registered will be 0.83 of the true mean.

Although the true e.m.f. can be obtained by the use of series or parallel couples, the translation of this into temperature does not necessarily give the true mean temperature, due to the relationship between them being a curve. An idea of the possible magnitude of the temperature error can be obtained by considering the case of two circuits. This will be done for copper-constantan.

In Fig. 27 T_1 and T_2 are the actual temperatures, e_m the observed mean e.m.f., which corresponds to the temperature T_m . T_a is the true mean temperature for which e_a would be the e.m.f.

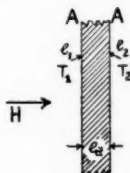


FIG. 28.

Taking the relationship as $e = aT + bT^2$, it is found that the error is given by

$$(e_m - e_a) = \frac{b}{4} (T_1 - T_2)^2$$

With normal copper constantan wires, the order of the errors is shown by the following table:

$T_1 - T_2$ deg. fahr.	Error in Deg. Fahr. with Mean Temperatures of		
	32 deg. fahr.	82 deg. fahr.	200 deg. fahr.
1	0.00016	0.00015	0.00013
10	0.016	0.015	0.013
20	0.060	0.057	0.050

Such errors are negligible and are greatly exceeded by observation and location errors.

The similar type of error in connection with the differential couples on the plates would be due to, say, a uniform temperature existing in a plate during calibration and an ununiform in application, although the same surface and differential values are obtained as before. The error for an extreme case of a difference between two halves of the plate of 20 deg. fahr. would be only 0.008 per cent.

APPENDIX C

Relationship of Differential to Surface E.m.f.

Let A , Fig. 28, be the section of a plate through which heat is passing at a constant rate

Let:

H = the rate of heat flow.

T_1 = hot side temperature measured above ice-point.

T_2 = cold side temperature measured above ice-point.

T_m = mean temperature measured above ice-point.

K = thermal conductivity coefficient, considered constant with temperature.

S = shape factor = reciprocal of thickness.

e_1, e_2 = E.m.f.'s of thermocouples corresponding to T_1 and T_2 respectively.

e_m = E.m.f. against the ice-point of the mean temperature of plate, that is $\frac{T_1 + T_2}{2}$.

e_d = Differential e.m.f. of the two surfaces, that is corresponding to $T_1 - T_2$.

All the e.m.f.'s are assumed reduced to the values given by single couples.

Then for the heat flow:

$$H = KS (T_1 - T_2) \dots \dots \dots (1)$$

The relationship between the temperature and e.m.f. of copper-constantan couples is given very closely by an equation of the form $e = aT + bT^2$, where a and b are constants and the temperature T the number of degrees above the freezing or ice-point.

From this it follows that:

$$T = \frac{a}{2b} \left\{ \sqrt{\frac{4b}{a^2} e + 1} - 1 \right\} \dots \dots \dots (2)$$

$$\text{also } e_d = e_1 - e_2 = a (T_1 - T_2) + b (T_1^2 - T_2^2)$$

$$= \{a + b (T_1 + T_2)\} (T_1 - T_2) \dots \dots \dots (3)$$

Let e_u be the e.m.f. units per unit of heat flow, then from (1) and (3)

$$e_u = \frac{e_d}{H} = \frac{a + b (T_1 + T_2)}{SK} = \frac{a + 2bT_m}{SK} = \frac{a}{SK} \sqrt{\frac{4b}{a^2} e_m + 1} \dots \dots \dots (4)$$

from substituting the value of T_m given by (2).

With a good grade of constantan wire, commonly used values for the constants are:

$$e = 21T + 0.0123T^2 \dots \dots \dots (5)$$

where e is in microvolts and T deg. Fahr. above 32.

Using these the portion of (4) under the radical has the following values:

e_m microvolts =	0	1000	2000	3000	4000	
$\sqrt{\frac{4b}{a^2} e_m + 1}$	=	1.00	1.052	1.104	1.156	1.208

Equation (4) can therefore be supplanted approximately by

$$e_u = \frac{a}{SK} \left\{ 1 + 0.000051 e_m \right\} \dots \dots \dots (6)$$

and have an error of only 0.2 per cent, and thus is practically a straight line.

With K constant the plotting of e_u , or differential microvolts per B.t.u., against the e.m.f. of the mean temperature should give a straight line, and little error would be introduced if the e_m were obtained from e_1 plus or minus $\frac{e_d}{2}$. For thin boards e_d is so small that it is negligible, so that the e_u could be plotted against e_1 or e_2 and still give a straight line within the range of experimental error.

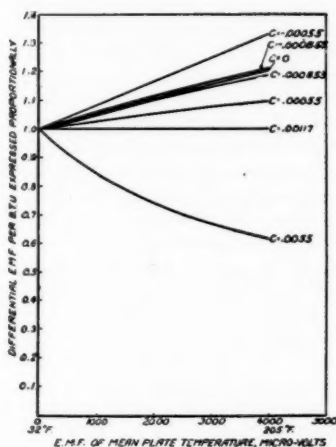


FIG. 29.

If K is variable, that is if the thermal conductivity changes with temperature, as it does for most materials, then it can be expressed as:

$$K = k(1 + cT_m) = k \frac{ca}{2b} \left\{ \sqrt{\frac{4b}{a} e_m + 1} - \left(1 - \frac{2b}{ca} \right) \right\}$$

Therefore

$$e_u = \frac{a}{Sk} \cdot \frac{\frac{2b}{ca} \sqrt{\frac{4b}{a^2} e_m + 1}}{\sqrt{\frac{4b}{a^2} e_m + 1} - \left(1 - \frac{2b}{ca} \right)} \dots \dots \dots (7)$$

When $\frac{2b}{ca} = 1$, $e_u = \frac{a}{Sk}$, that is e_u is constant. This gives a value of $c = 0.00117$ when a and b have the values used above.

Fig. 29 shows curves with several assumed values of c . All are approximately straight lines within the limit of $e_m = 4000$ microvolts.

If due to the shortness of the differential couple wires, their point temperatures are not the same as the surface of the plate, yet, as shown in Appendix A, their

actual temperature differences will always be proportional to the plate temperature difference. It therefore follows that the slopes of the curves will not be affected, and this slope depends only on the values of a , b and c .

APPENDIX D

Air Infiltration Through a Porous Wall as Affecting the Heat Transmission

The assumption is made that the material has fine pores uniformly distributed, and the air passing through is thus uniformly distributed. Also that the rate of passage is sufficiently slow so that the air will take the temperature of the material with which it is in contact.

The treatment given here will be brief, and the further assumption made that the gas at the entering face has the same temperature as that wall face. For small flows this assumption will not be far wrong.

Let Fig. 30 show a section of the wall, the air passing from the hot to the cold side. The temperatures are in degrees fahrenheit, and thicknesses in inches. Considering 1 sq. ft. of the area, let $g = wc_p$, where w = lb. of air passing per sq. ft. per hr., and c_p = specific heat of air at constant pressure.

Let H be the B.t.u. per sq. ft. per hr. entering the wall face. This does not include that contained in the air entering, but is the same type of flow as occurs without the infiltration, and this same heat can be considered all through the wall as independent of the heat in the air.

Since the gas enters with temperature T_1 and at a distance x has a lower temperature T , it has given up an amount of heat $g(T_1 - T)$. The total heat passing across the plane at x is thus $H + g(T_1 - T)$. It follows that if the thermal conductivity coefficient of the material at the temperature T is $(a + bT)$,

$$H + g(T_1 - T) = -(a + bT) \frac{dT}{dx}$$

Integrating this for the whole wall gives

$$\log \left\{ 1 + \frac{g}{H} (T_1 - T_2) \right\} = \frac{gX + b(T_1 - T_2)}{a + b \left(\frac{H}{g} + T_1 \right)} \dots \dots \dots (1)$$

If the air is passing in the other direction, that is from the cold to the hot side, equation (2) still holds, if g be replaced by $-g$.

The equations in this form are of interest in connection with furnace walls, in which the variation of conductivity with temperature is large, but for ordinary temperatures it may be considered constant and $(a + bT)$ replaced by k . With air passing from the cold to hot side this gives:

$$H = \frac{g(T_1 - T_2)}{1 - e^{-\frac{gX}{k}}} \text{ where } e = 2.718 \dots \dots \dots (2)$$

also temperature T at x is given by,

$$T = T_1 - (T_1 - T_2) \frac{1 - e^{-\frac{gX}{k}}}{1 - e^{-\frac{gX}{k}}} \dots \dots \dots (3)$$

From equation (3) the temperature curve through the wall can be plotted. The H , equation (2), is the total heat lost from the room by conduction. In addition

to pounds of air at a temperature T_1 enters the room and this brings up the interesting problem as to how it should be considered. For small infiltration it will be equivalent to being part of or increasing the convection factor of the surface transmission coefficient.

APPENDIX E

Publications Referred to in the Paper

- (1) Transmission of Heat Through Heavy Building Materials, by A. H. Barker, University of London, Department of Heating and Ventilating, Bulletin No. 2, 1918. It gives a very clear exposition of the principles and records the first attempt made to measure the heat flow through a portion of a wall, as well as describing the laboratory method used. Apparently a Bulletin 2A has been issued.
- (2) Coefficients of Heat Transmission Through Heavy Building Materials Approved by the British Institute of Heating and Ventilating Engineers. Proceedings V., xlx, 1920-21, p. 228. This is a preliminary publication. Values are based on tests at the University of London.

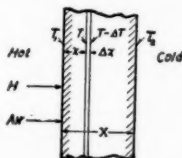


FIG. 30.

- (3) Heat Loss Through Various Types of Building Construction, by L. A. Scipio. JOURNAL OF THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Vol. 27, Sept. 1921, p. 637. Review of results on five test houses erected in northern Minnesota. This investigation was not carried very far.
- (4) Building Wall Construction in Wood and Brickwork. Results of experiments at the Norwegian Technical Academy, by Professor A. Bugge. Tehnisk, Ukeblad, Nos. 43 and 44, 1921. Record of results of 24 experimental dwelling houses of wood, brick, tile and cement, which were maintained at an approximately constant internal temperature during one and one-half winters. The houses were complete with cellars, and the relative construction costs are given. A full record of this work will be published in English at an early date.
- (5) Heat Transmission Through Building Walls, by Alf Kolflaath, Teknisk Ukeblad, Nos. 38, 39 and 41, 1923. Record of results on an analysis of some of the houses of reference 4 by the use of Barker's Box method.
- (6) Investigations Relating to the Heat-Insulating Efficiency of Building Construction, by H. Kreuger and A. Eriksson, Academy of Engineering Science, Proc. No. 7. Published by The Gunnar Tisell Tech. Pub. Co., Stockholm. Issued, 1922. Record of results of their tests to that time. The transmission through windows is treated very thoroughly as well as through a variety of wall sections. The tests were on the principle of Barker's box method, the walls under tests forming a removable panel of an enclosed chamber within another one, the interior of the first being kept at low temperature, and the exterior at room temperature. This work is still proceeding.

- (7) The Transmission of Heat by Radiation and Convection, by Ezer Griffiths and A. H. Davis. Department of Scientific and Industrial Research; Food Investigation Board Report, No. 9, 1922. Although this investigation was mainly for the benefit of the refrigerating industry it is equally useful in connection with the surface coefficients of structures. Deals with transmission across air spaces and from walls of various heights.
- (8) Heat Transmission Through Walls, Concretes and Plasters, by Ezer Griffiths, Department of Scientific and Industrial Research; Building Research Board Report, No. 7, 1923. Cover tests using the hot plate method on 15 types of walls, and 11 plasters. This and No. 7 are obtained from H. M. Stationary Office, Imperial House, Kingsway, London W. C. 2, England.

DISCUSSION

L. A. HARDING: The Research Laboratory is to be congratulated upon the production of such a remarkably accurate and sensitive piece of apparatus. Personally, I think the time is about right for the Society to adopt a standard method for determining heat transmission and I would like to recommend, that the Research Laboratory be called upon to present to this Society at its next meeting, a code which in their opinion is best adapted for this purpose.

M. S. VAN DUSEN: I would like to reiterate the statement of the previous speaker to the effect that Mr. Nicholls ought to be congratulated upon his work in actually obtaining the calibration of these plates.

In dealing with a very thin plate such as that described, there is some possibility of certain errors being introduced, when the plate is actually applied to the wall. When a plate is applied to a wall, there is a certain amount of resistance between the wall and the plate. However, if the resistance is uniform over the surface, it will be all right. If, on the other hand, it is not uniform, there is a slightly greater contact effect at one place than another. The average temperature of this surface of course is given by a large number of thermocouples but the temperature distribution over the plate is disturbed to a certain extent and the heat no longer flows perpendicular to the conductimeter, as it does during calibration. Therefore, in actual operation we may have heat flowing in one portion in one direction, in another portion in another direction and our calibration will not hold for these conditions.

F. G. HECHLER: For several years members of the A. S. H. & V. E. have cooperated with, and been interested in, the investigations in heat transmission at the thermal plant of the Pennsylvania State College, and it has been suggested that the Society as a whole would be interested in a brief statement of the progress of these investigations.

The construction of the present heat meters and their application in both the laboratory and on the walls of existing buildings is the outgrowth of an interesting series of investigations covering some 10 years. The original heat transmission laboratory was built in 1910 from designs prepared by L. A. Harding who was then Professor of Mechanical Engineering at Penn State.

Early reports showed that the box method is open to severe criticism for general use. When this became evident plans were made by the Engineering Experiment

Station for the construction of a guarded hot plate. This plate was designed with an effective heating element 2 ft. square surrounded by a 6 in. guard ring which permits the use of test specimens 3 ft. square. A preliminary report by Houghten and Wood describing the construction and calibration of this plate appears in the July, 1921, A. S. H. & V. E. JOURNAL.

The conclusion was reached that the "hot plate" possessed many advantages over the box method for determining wall conductivity, so the hot plate was discarded for the measurement of heat flow in a wall and standard resistance plates or "heat meters" were substituted; with the result that conductivity tests made by the "hot plate to air" method have given more consistent results than those previously obtained by other methods.

With the excellent facilities available it was decided to use the "hot plate" method for the calibration of the standard "heat meters." In order to determine accurately the heat distribution on the two sides of the hot plate it is necessary to use four resistance plates, two on each side of the hot plate.

The calibration of four $\frac{1}{2}$ in. cork heat meters is virtually completed. Check tests made months apart have given consistent results showing the permanency of the plates. The calibration of four additional plates, two of Ebony Asbestos Wood and two of Transite, is now under way. The application of heat meters for laboratory use is also being tested in connection with materials of known conductivity against a tank containing crushed ice.

The investigations have been going forward steadily, although many interruptions have interfered with the calibration of these plates. Present progress of the work indicates that the results will be available within a few months.

C. H. HERTER: The paper reveals a vast amount of research work in the field of heat transmission. A very ingenious heat transmission meter is described which should facilitate heat flow measurements. A similar plate was described by Houghten and Wood in the July, 1921, and March, 1922, issues of the JOURNAL. This being a new development it will take sometime before the apparatus is perfected, so that it will meet all requirements.

The plates appear to be too good, in that they are too sensitive. Thus from Fig. 21, on a 24 in. brick wall, the heat flow seems to fluctuate to such an extent that in 60 seconds it changes from 7.7 to 8.2 B.t.u., which as measuring a flow through the whole wall would necessitate a change of 3 deg. between surfaces. It therefore seems to indicate some variations occurring at the room side of the wall when it should show the actual heat flow through the wall.

In an article, Improvements in Thermometry, (*Refrigerating World*, Feb., 1923) by the writer, a transmission meter designed by E. Smidt is described. It consisted of a 4 in. square of glass, $\frac{1}{8}$ in. thick with both faces covered with nickel foil. This meter was used at the Munich Technical High School against a 1 year old brick wall, giving a heat conductivity of 3.71 B.t.u.

Glass would seem to be a desirable material for such plates because it is homogeneous, air and moisture proof, and permanent. If grooving and drilling for wires is necessary, this can be done and the glass annealed so as to relieve all strains.

Those interested in research should examine at least the article on Transmission of Heat through Building Constructions, by Prof. O. Knoblauch, E. Raisch and H. Reicher in *Gesundheits-Ingenieur*, vol. 43, no. 52, Dec. 25, 1920, also a 124 page book, Die Waerme Verluste Durch Ebene Waende (Heat Losses through Building

Walls), by Dr. Karl Hencky. The Germans built sample walls which were weighed and took six months to dry out, and then took 16 to 52 days per test to secure constant conditions and reliable results. They were not satisfied until the results by different methods agreed. From the tests it was possible to secure the necessary fundamental data, as given in Hencky's book, by means of which the heat flow can be calculated to the same degree of accuracy as may be obtained from tests direct.

P. NICHOLLS: Replying to Mr. Herter, the varying flow indicated by the plates is not an error, but merely shows that the rate of heat flow into a wall is not constant under natural air exposure. Under variable conditions it is incorrect to speak of a given value as indicating the instantaneous flow through a wall, since the rate of flow will vary at different sections of the wall, as is shown in Fig. 2-C. A meter plate will indicate the rate of flow at the position it occupies, and plates put on opposite faces of a wall have given flows in contrary directions as would be required by Fig. 2-C.

Glass in itself would be a good thermal resistance material, but would be impossible to use in large sizes and with a large number of couples.

W. H. CARRIER: I would like to ask Mr. Nicholls if the meter could be applied to a ceiling, for example, to be used to insulate against sunlight, if we had a concrete roof. You know that when the sun is shining on it the temperature may vary from 120 to 150 deg. in the summer time. Would it be possible to get the correct transmission of that roof during the period of sunlight and also some method of determining the lag of heat transmission similar to the method he showed? Would it be possible to make an analysis, in other words, what actually took place and then predetermine in other cases what might be expected to take place with a roof of somewhat different construction?

P. NICHOLLS: I regret that the paper was so long that it was impossible to cover everything. Most of the questions you will find answered in the paper particularly in regard to the applications brought up by Mr. Carrier. When you put the plate on a wall all the plate does, is to give the heat flow into the surface from moment to moment as long as you like, thus you get a complete record of the heat flow.

Dr. Van Dusen's criticisms, are of course possibilities, although in the testing of the plates, they were assembled in a bunch and each time I set them up I did not get the same exact contact. Sometimes they would be tight at the bottom and sometimes I had actual air gaps at one side or the other. When I put air gaps between thin plates they did not lie exactly parallel and one part would have a bigger bulge, and yet you can see from the paper the variations I got were extremely small in percentage.

F. PAUL ANDERSON: I would like to make one comment. The essential point to bear in mind in reference to this heat meter is that it measures accurately the heat passing through.

L. A. HARDING: Do we understand this meter is a meter for the conductivity of the material only, or are we to understand it is a meter that measures the heat passing through a wall from air to air?

P. NICHOLLS: The plate measures the rate of heat flow at the point and if we want to derive any constant from it, we have to take our observations over a long enough time to get a summation.

No. 686

AIR LEAKAGE THROUGH THE OPENINGS IN BUILDINGS

By F. C. HOUGHTEN¹ AND C. C. SCHREADER,² PITTSBURGH, PA.

MEMBERS

HEAT is lost from buildings in two ways: *First*, by transmission and *Second*, by infiltration. Both sources of heat loss are of vital concern to the heating and ventilating engineer, the architect and the owner. Both are difficult of exact measurement and determination of constants which may be used in practice with the desired engineering accuracy. As a result, the calculation of heat loss from buildings probably involves a greater element of chance than any other engineering problem.

Heat loss by transmission was one of the first problems to receive the attention of the Research Laboratory. Total heat loss by infiltration for a room as a unit has also received considerable attention. (JOURNAL AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, January and September, 1921.) In January 1916, a paper on Window Leakage by Stephen F. Voorhees and Henry C. Meyer Jr., was presented at the Annual Meeting of this Society (TRANSACTIONS, A.S.H. & V.E., 22, 1916, p. 183).

The great need for information regarding infiltration led to the present investigation of the leakage of air through and around all types of windows and doors by the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, in cooperation with the *American Institute of Architects* and the U. S. Bureau of Mines. The architect is interested in the relative leakage of air through various types of windows and doors, with and without weatherstripping, in order that he may design a building with the lowest heat loss consistent with other considerations. The heating engineer needs the actual leakage through and around all types and sizes of windows and doors, or better, through a unit length of crack around such openings, in order to more accurately calculate the heat loss from any room or building and supply radiation accordingly.

This report deals with the method employed in the investigation of and results obtained for double hung windows, 2 ft. 8 in. x 5 ft. 2 in. x 1³/₈ in., in a 13 in. brick wall, plastered on the inside with cement plaster. Results are given for the leakage,

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² U. S. Bureau of Mines.

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through such a window without weatherstripping with two types of weather stripping, around the frame, and through the brick wall itself.

Leakage of air through cracks around windows and doors, cracks in walls, and through the porous materials of which walls are made, takes place in accordance with two physical laws. *First*, there is an interchange of air through the wall by diffusion; *Second*, there may be a current of air through the wall caused by a pressure difference set up by the impinging wind. The first goes on at all times, is independent of wind velocity, and is probably negligible. The second takes place only when there is a pressure difference between the two sides of the wall. Such a pressure difference exists whenever the wind blows against the surface of the wall or whenever the direction of the wind toward the wall is changed. For any given velocity of wind striking the wall at right angles, there is always a definite pressure produced at the surface which tends to cause leakage of air through cracks.

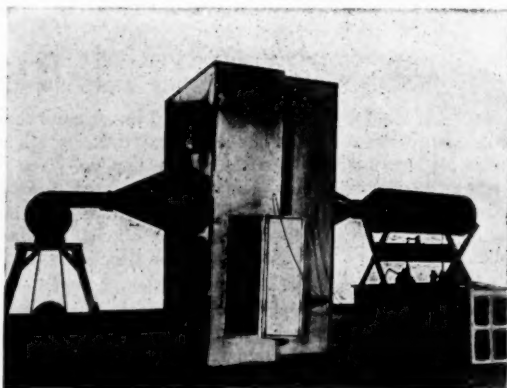


FIG. 1. APPARATUS FOR TESTING WINDOW LEAKAGE

The amount of air leakage for any crack for a given pressure difference is the same regardless of whether this pressure is produced by wind or some other cause.

Uniform air velocities over a large area for a long period of time are hard to produce and difficult to duplicate. It is much easier to produce and duplicate pressure differences on the two sides of a window by means of a blower. It was, therefore, decided that for this investigation the apparatus should be so designed that a blower could be used to produce a pressure drop through the test window built in a section of wall.

Apparatus

After carefully considering all phases of the problem, the apparatus shown in Fig. 2 was designed by and built under the direction of the Research Laboratory. In many respects it is similar to the apparatus used by Voorhees and Meyer in the work previously mentioned. The apparatus is built of 18 gage galvanized iron, and consists of a pressure chamber *A* and an air collecting chamber *B* separated by a section of wall including the particular window or door to be tested. Air pressure is produced in the first chamber by means of a motor driven blower, and the volume of air passing through the wall is measured by the orifice box *C*. The test wall,

10 ft. high x 6 ft. 6 in. wide, is built in the collecting chamber section flush with its outer edge and the pressure chamber section of the apparatus bolted on later. The desired pressure is produced in *A* by varying the inlet of the blower, and by means of a butterfly damper and relief slide in the connection between the blower and *A*. Chamber *A* is substantially air-tight although the requirements of the investigation do not demand that it be absolutely so. A door, 4 ft. x 1 ft. 6 in., allows entrance into this chamber to make any changes in the opening under study. The present blower has a capacity of 1100 cu. ft. per min., at 5 in. water pressure. Chamber *B* must be air-tight so that all air passing through the test wall

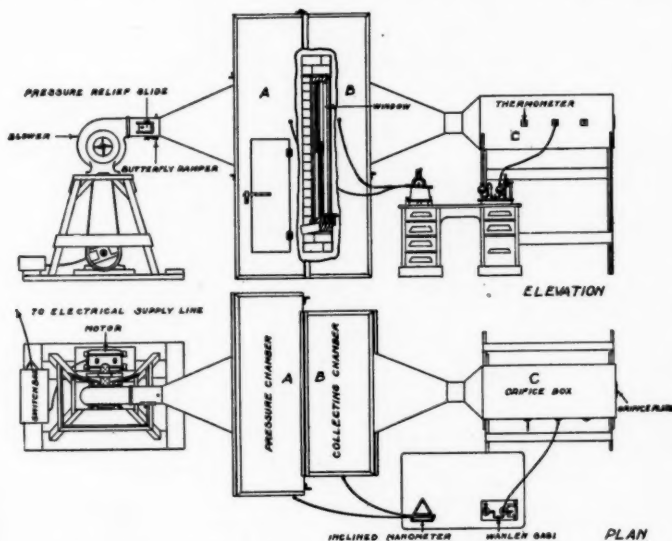


FIG. 2. DIAGRAM OF APPARATUS FOR TESTING WINDOW LEAKAGE

must pass through the orifice box used for measuring its volume. Every precaution, including soldering and painting the joints, was taken to make this part of the apparatus tight. Tests which will be described later in the report show that this condition was practically obtained.

The orifice box is one used by the Bureau of Mines for measuring the flow of steam and air in connection with boiler tests. The box is cylindrical in shape, 24 in. in diameter, with the orifice plates in the end. Orifice plates with openings varying from $1\frac{1}{32}$ in. to 5 in. in diameter were made so that they were easily interchangeable. These were carefully turned out in the instrument shop of the Bureau of Mines in accordance with R. J. Durley's specifications. The law of the air flow through orifices has been well established by Durley (*A.S.M.E. Transactions*, Vol. 27, p. 193) and others, and is given by the equation:

$$Q = 1096.5 C A \sqrt{\frac{p}{w}} \quad (1)$$

Q = quantity of air, cu. ft. per min.

A = area of orifice in sq. ft.

p = pressure head in inches of water causing flow through the orifice

w = weight of air in pounds per cu. ft.

C = coefficient of discharge

The coefficient of discharge used is 0.6 because it approaches this value at the pressures obtained for all the orifices used.

The pressure drop through the orifice which in this case is the difference between

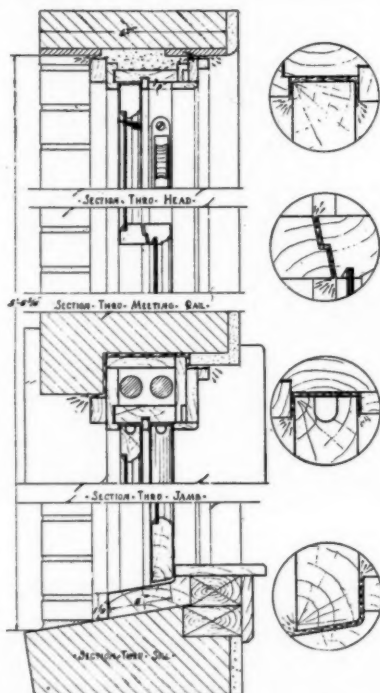


FIG. 3. DETAILS OF WINDOW WITHOUT WEATHERSTRIPPING

the pressure in the orifice box and the atmosphere, was measured by a Wahlen gage accurate to 0.0001 in. of water. This gage was developed at the University of Illinois and is fully described by A. C. Willard in the University of Illinois Engineering Experiment Station Bulletin 112.

While the accuracy of the orifice method of measuring air flow is well accepted by those familiar with its use, it was thought desirable to compare it with some other method. The orifices in the box as used in the tests were compared with a dry gas meter used as a standard in the meter testing laboratory of the Equitable Gas Co., Pittsburgh. These tests showed that the results for the orifices using the equation

given above were more consistent than those for the gas meter with which it was compared. As a further check of the relative readings of the various sized orifices they were compared with each other and with duplicate orifices by placing a second orifice in the window opening in the test wall in series with the box. This was done when the total leakage through the wall was reduced to a negligible but known value.

The pressure drop through the test wall was measured by an inclined U-tube gage of a particularly accurate type designed and built by the Bureau of Mines.

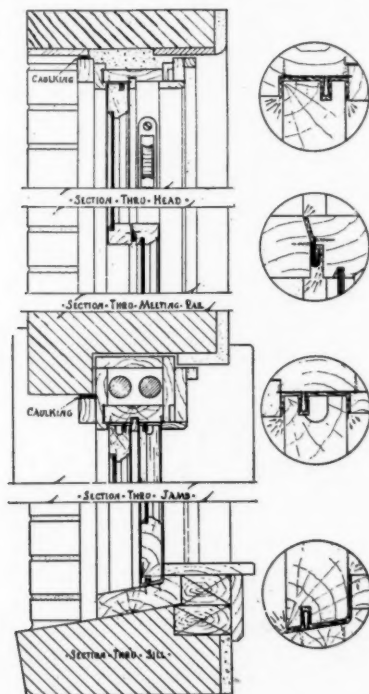


FIG. 4. DETAILS OF WINDOW WITH RIB TYPE WEATHERSTRIPPING

It was compared with the Wahlen gage and found to be accurate to 0.003 in. of water. The two legs of this gage are connected by rubber tubing to chambers A and B.

A test of any particular window was made by regulating the blower pressure so as to give the desired pressure drop through the window indicated by the differential gage. The size of the orifice chosen for any test was such as to give a pressure in the orifice box of from 0.3 to 0.7 in. of water. When these conditions were maintained for a sufficient time to insure equilibrium, the two pressure gages were read simultaneously. By repeating the tests for a large number of pressure differences

through the window, data was obtained for plotting a curve giving leakage through the wall in cu. ft. per min. against pressure differences in inches of water or wind velocity. All velocities and volumes given are for air weighing 0.07488 lb. per cu. ft. corresponding to air having a barometric pressure of 29.92 in. of mercury, a dry bulb temperature of 68 deg. fahr. and 50 per cent relative humidity. Many

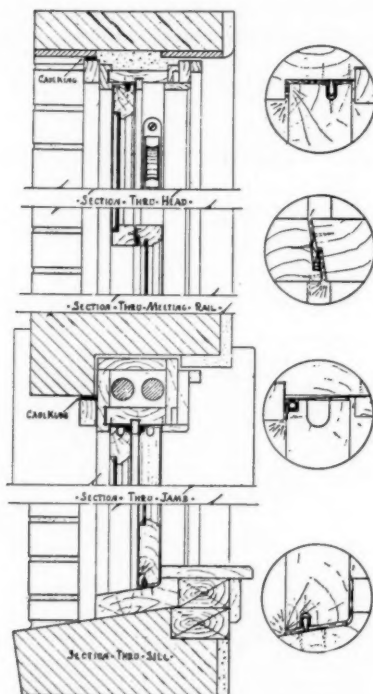


FIG. 5. DETAILS OF WINDOW WITH INTER-LOCKING TYPE WEATHERSTRIPPING

tests in such a series were repeated after opening and closing the window and also after taking the sash out of the frame and replacing it.

Data and Results

The results given in this paper are for a double hung window, 2 ft. 8 in. x 5 ft. 2 in. x $1\frac{3}{8}$ in. in a 13 in. brick wall plastered on the inside with cement plaster. The brick wall was built, the plastering was done and the window hung by mechanics in the employ of large contracting firms in the city of Pittsburgh. The work was done according to specifications supplied by and under the direction of S. F. Heckert, representing the Structural Service Committee of the American Institute of Architects. All changes in the window, such as hanging new sash and applying weatherstripping, were made also by mechanics under his direction. Every precaution

was taken to make the wall and window represent work done by the ordinary contractor under the supervision of an architect. The sash and frame were given three coats of paint. Fig. 3 is a vertical section through the unweatherstripped window with a horizontal section through one side of the frame.

For convenience in presenting, the results are given in two sections. *First*, those obtained in a preliminary series of tests on the unweatherstripped window in the wall as built, and with certain changes such as calking the frame, sealing cracks, and painting the wall; *Second*, results obtained from a large number of tests with various sash hung under different conditions with and without weatherstripping.

Preliminary Tests

Preliminary tests were made in order to study the working of the apparatus itself and in order to differentiate between the various channels of leakage through the

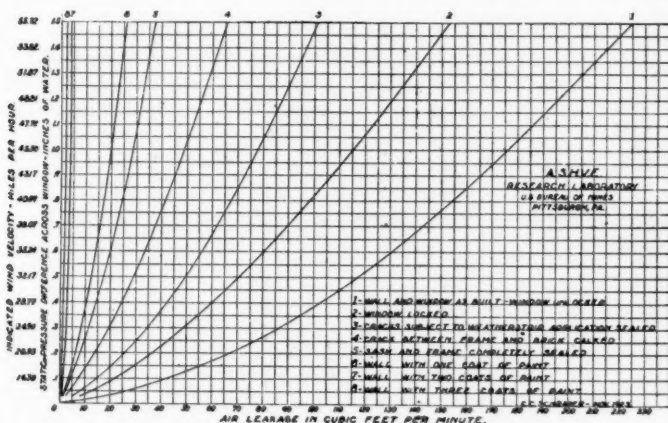


FIG. 6. RESULTS OF TESTS OF LEAKAGE THROUGH VARIOUS PARTS OF WINDOW AND WALL

window and wall. Leakage through the window may be divided into the following parts. *First*, that which passes through the cracks, around the sash perimeter which are subject to weatherstrip application; *Second*, that which passes through the cracks between the frame and the brick and can be eliminated by calking under the staff bead or brick mold. This may be called the frame leakage. *Third*, leakage through other cracks in the frame or sash which cannot be eliminated by either weatherstripping or calking and may be called the "elsewhere" leakage.

Before making the first series of tests, the joint between the brick and the chamber wall was calked so that all leakage would take place through the wall or window. In all other respects, the wall and window were in the condition in which they were left by the mechanics, the sash having been fitted as tight as would allow free sliding, though probably tighter than would be allowable in actual construction because of swelling in rainy or damp weather. The window was left unlocked. A large number of tests were made with various pressure drops through the wall, many of them being duplicated several times after opening and closing the

window, in order to determine the variation due to the way in which the window was closed. No care was taken to close the window in any particular way other than to see that the lower sash was pushed down against the sill and the upper sash raised until the meeting rails were even. Curve 1, Fig. 6, shows the leakage for this condition for various pressures or wind velocities. The shape of the curve is characteristic of all curves obtained with the various conditions of the window and, as would be expected, shows the same characteristics as the curve for the flow of air through an orifice. For a pressure difference of 0.1 in. of water through the wall corresponding to a wind velocity against the wall of 14.4 miles per hr., 42 cu. ft. of air per min. passed through the window and wall. With a pressure drop

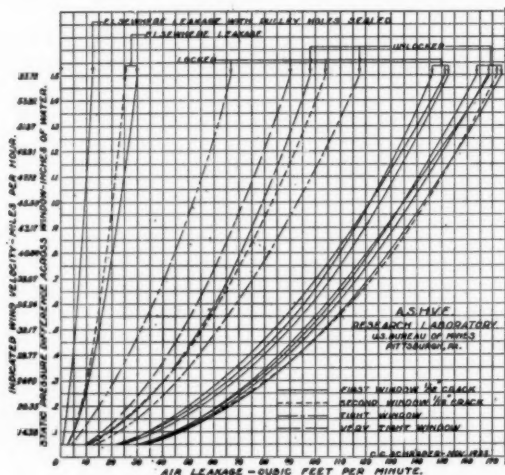


FIG. 7. RESULTS OF TESTS ON WINDOWS WITH VARIOUS CRACKS, SHOWING VARIATION IN LEAKAGE FOR DIFFERENT TESTS ON SAME WINDOW

through the wall of 1 in. of water, corresponding to a 45.5 mile wind velocity, 174 cu. ft. per min. passed through.

The second series of tests was made under the same conditions as the first series excepting that the window was locked. Curve 2 shows the leakage for various wind velocities for the locked window. Locking caused a reduction in leakage of 20 cu. ft. per min. with a 14.4 mile wind and 64 cu. ft. per min. with a 45.5 mile wind. The third series of tests was made with the cracks around the sash perimeter, which are subject to weatherstrip application, sealed with a rubberized adhesive tape. This tape was tested and found to be as effective as a plastic calking compound and was more easily and quickly applied and removed. The leakage for this series of tests is given in Curve 3, and the difference between this curve and Curve 1 or 2 indicates the maximum possible reduction in leakage by a perfect weatherstrip.

Before making the next series of tests the staff bead, or brick mold, was removed and the crack between the frame and the brick wall calked. The brick mold was then replaced. Calking was also applied between the frame sill and the brick. The leakage for this condition is given in Curve 4 and the difference between Curve

4 and Curve 3 gives the leakage between the frame and the wall, commonly called the frame leakage.

In order to determine the elsewhere leakage, a sheet of galvanized iron was fastened by screws over the entire frame and the edges were sealed with calking compound. The leakage for this condition is given in Curve 5. The difference between Curve 4 and Curve 5 is the leakage stopped by the galvanized iron and is the elsewhere leakage.

Curve 5 shows a considerable leakage which does not go through the window opening, but through the brick wall and the plaster. To prove that this leakage was really through the brick wall, the wall was painted one coat with asphaltum paint and another series of tests made. The result of this series is shown in Curve 6. The difference between Curves 5 and 6 represents the leakage stopped by one coat of paint. The wall was then thoroughly inspected and any visible cracks in the mortar closed with calking compound and given second and third coats of paint after each of which additional series of tests were made resulting in Curves 7 and 8, respectively. These curves show the reduction in leakage through the wall by each coat of paint. Another coat of paint was applied later and the leakage through the wall was further reduced

to one half of that shown in Curve 8. The total leakage through the entire wall had been reduced by the various processes from 4.5 cu. ft. per min. to 0.2 cu. ft. per min. for a 14.4 mile wind, and from 28 cu. ft. per min. to 0.9 cu. ft. per min. for a 45.4 mile wind. No doubt further painting would have reduced the leakage still more, but that shown by Curve 8 was so small that it was considered negligible.

With the leakage through the window and wall reduced to a minimum, some special tests were made in order to determine the magnitude of any leakage which might occur from chamber B. The leakage through the wall and window as indicated by the orifice reading is too small by the amount of the leakage from chamber B. While every precaution was taken to eliminate this leakage, it was not possible to do so entirely. However, as shown by the following tests, it was negligible.

When the leakage through the wall as shown by the orifice reading was reduced to a minimum, a pressure drop of 1.5 in. of water through the wall, gave a pressure difference of 0.066 in. between the second chamber or orifice box and the atmosphere when a $\frac{1}{8}$ in. orifice was used. That is, 1.41 cu. ft. per min. passing through the orifice and an unknown amount which we will call x was leaking from

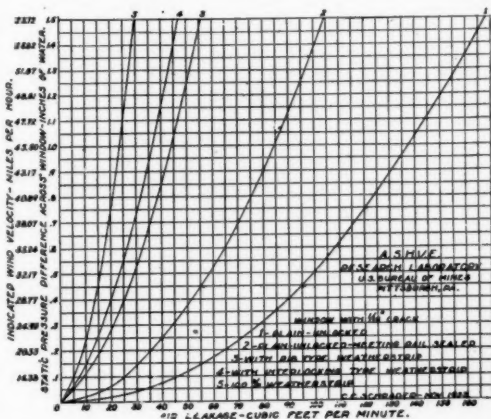


FIG. 8. RESULTS OF TESTS ON WINDOW WITH $\frac{1}{16}$ " CRACK AROUND PERIMETER

the second chamber. The leakage through the wall was then $1.41 + x$ cu. ft. per min. We wish to determine the value of x for all pressures. Since x cu. ft. per min. were passing through minute openings with an orifice pressure p , x is given by the orifice formula as:

$$x = 1096.5 C A \sqrt{\frac{p}{w}} \quad (2)$$

where the various symbols have the same significance but probably not the same values as given in equation (1). A and C are not known but are constant for the

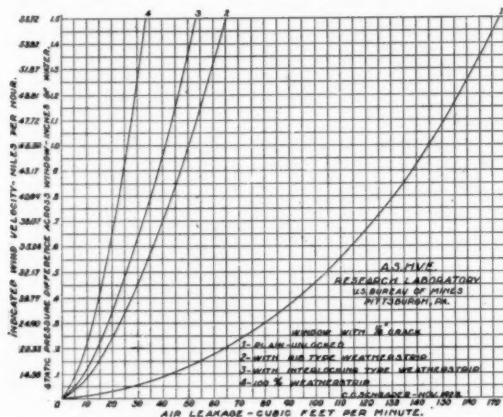


FIG. 9. RESULTS OF TESTS ON WINDOW WITH $1/8$ " CRACK AROUND PERIMETER

same conditions and w is also constant; $A C$ and w can therefore be included with the numerical constant 1096.5 as K . Our equation then becomes,

$$x = K \sqrt{p} \quad (3)$$

and the leakage from the second chamber for an orifice pressure of 0.066 in. becomes:

$$x = K \sqrt{0.066}$$

The leakage through the wall for any pressure drop may likewise be expressed as:

$$y = K_1 \sqrt{p} \quad (4)$$

and for a pressure drop of 1.5 in. as,

$$y = K_1 \sqrt{1.5} = 1.41 + K \sqrt{0.066} \quad (5)$$

The orifice was eliminated by using a plate without a hole and the leakage through the wall became equal to that from the second chamber. The pressure drops observed through the wall, and between the second chamber and atmosphere were 0.045 in. and 0.701 in., respectively; therefore,

$$\begin{aligned} y &= K_1 \sqrt{0.045} = x \\ &= K \sqrt{0.701} \end{aligned} \quad (6)$$

Solving equations (5) and (6) simultaneously gives $\bar{K} = 0.308$ and the leakage from the second chamber for all pressures becomes,

$$Q = 0.308 \sqrt{p} \quad (7)$$

This gives a leakage from the second chamber of 0.258 cu. ft. per min. for an orifice pressure of 0.7 in. of water, the maximum used in the tests. This leakage is entirely negligible in comparison with the results obtained.

When the galvanized plate and the tape were removed from the sash perimeter cracks, tests were made in order to check the decrease in leakage resulting from their application. The calking around the frame and the paint on the wall was not removed after having been applied, so that the curves in all figures after Fig. 6 do not include the frame and wall leakage, and show only the leakage through the window.

Tests on Windows with and without Weatherstripping

After completing the preliminary series or tests, a large number of tests were made with a number of sash with and without weatherstripping and with various width of crack around the sash perimeter. As was mentioned before the preliminary tests were made with a sash too light for practical purposes. Tests were made with cracks of $1/16$, $1/8$, $3/16$ and $1/4$ in. on both sides, top and bottom of the sash, without weatherstripping and with two types of weatherstripping. The size of the crack was increased to approximate the condition of windows that become loose, as is found in old buildings.

In these tests the sash were often changed and at least two different sash were fitted and tested for each condition. Figs. 4 and 5 show vertical sections of the window with the two types of weatherstripping together with horizontal sections through one side of the sash and frame, and also detailed sections of the various weatherstripped cracks.

The curves in Fig. 7 show the variation in data obtained for different windows fitted in the same way, for the same sash removed and replaced several times; also the leakage for tight windows and the effect of sealing the pulley holes. The five curves for the unlocked window with $1/16$ in. crack show the variation which can be obtained for the same window under different conditions and for a second window fitted as nearly the same as could be done by a carpenter. The greatest variation from the mean of the five series of tests is about four per cent. The variation in the leakage of the same window locked shows the effect that locking may have. The main effect of locking is on the leakage through the meeting rail crack. The lock on the sash giving the three solid line curves was put on by the carpenter in the usual manner. The lock on the sash giving the curve with the short dashes was put on by a member of the laboratory staff in such a manner as to draw the meeting rails together as tightly as possible. The locks on the weatherstripped windows were put on by the carpenter. Locking caused no reduction in leakage for these windows.

The tight window was fitted so as to allow opening without great difficulty. The very tight window required considerable effort in opening.

In the tests for "elsewhere" leakage the cracks around the upper sash were sealed on the inside because the weatherstripping was put on the inward side of the pulley holes and thus would not reduce the leakage through these holes. The cracks around the lower sash were sealed on the outside because the weatherstripping was applied near this side of the sash. A series of tests was made in order to determine the percentage of the elsewhere leakage which passed through the outer pulley

holes into the weight box and out through the inner holes or through cracks in the frame. Curves in Fig. 7 show that more than half of the elsewhere leakage occurred through these holes.

Figs. 8 to 11 give the results for the various sized cracks without weatherstripping, with two types of weatherstripping, and with 100 per cent weatherstripping, that is, with the cracks subject to weatherstrip application sealed up thus allowing only the "elsewhere" leakage. In each case the curve given is the average of several tests.

The curve for the unlocked window with the $\frac{1}{4}$ in. crack as obtained from the test data shows less leakage than the same condition with smaller cracks. While this is contrary to what might be expected it can be explained and corrected as outlined in the following paragraphs.

In testing the windows without weatherstripping, the lower sash was pushed down against the sill and the upper sash raised until the meeting rails were even. Be-

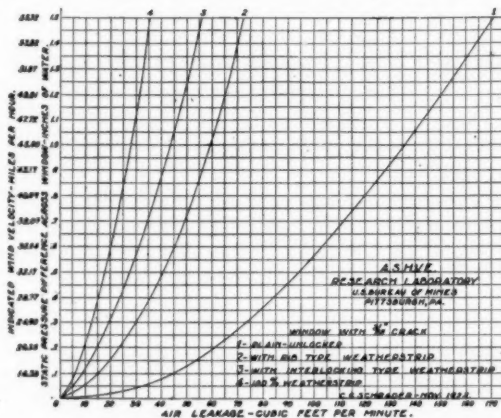


FIG. 10. RESULTS OF TESTS ON WINDOW WITH $\frac{3}{16}$ " CRACK AROUND PERIMETER

cause of the construction of the meeting rails, raising the upper sash beyond this point would reduce the crack between them. With the $\frac{1}{4}$ in. crack and the meeting rails even, it was found that the upper sash would just come up to the edge of the outside stop on the head of the frame. This stop extended $\frac{1}{2}$ in. from the frame. Also, if the sash were not planed off parallel to the head stop, there would be a visible crack. In order to get a test, the conditions of which would compare with the conditions of the preceding tests, the upper sash was raised until it was above the edge of the head stop. By doing this, the crack between the meeting rails was decreased and the resultant leakage was less than that for the smaller cracks. In order to correct for this decrease, a series of tests was made with the crack between the meeting rails sealed. The results of this series are shown in Fig. 11. Curve 2 shows the leakage with the crack between the meeting rails not sealed and Curve 3 the leakage with the same crack sealed. The windows with the $\frac{1}{16}$ in., $\frac{1}{8}$ in., and $\frac{3}{16}$ in. cracks were then tested with this crack sealed to determine the leakage between the meeting rails with the members even. The leakage thus found, sub-

tracted from that found without this crack sealed, represents the leakage through it. This proved to be much greater than that found in the tests on the window with $\frac{1}{4}$ in. crack with the meeting rails uneven. Also it was found to be practically the same for all three windows. An average was taken and the leakage for the window with $\frac{1}{4}$ in. crack corrected accordingly. Curve 1, Fig. 11, shows the corrected values. The difference between Curves 1 and 2, Fig. 8, shows the leakage between the meeting rails for the window with $\frac{1}{16}$ in. crack. These tests also showed that the leakage for all the windows with the crack between the meeting rails sealed was practically the same.

An examination of the curves for a plain window with different size cracks shows only a small variation for the three smaller ones. Various factors must be taken into consideration to account for this. The thickness of the sash is $1\frac{1}{4}$ in. and it slides in a $1\frac{1}{16}$ in. groove. If the sash were held in the middle of this space between the stops there would be a crack $\frac{1}{32}$ in. wide on either side. In this position the smallest crack around the edge of the sash through which air must pass is a maximum. The moment the wind strikes the window it tends to move it against the inside stops, thus increasing the crack on the outside but decreasing the crack on the inside. Since leakage depends largely upon the minimum width of crack around the sash perimeter, it is limited by the tightness with which the sash is forced against the stops. Increasing the width of crack around the edge of the window does not increase the minimum crack width and hence the leakage is not increased measurably.

When weatherstripping is used the window is held in the middle of the groove. The cracks between the members are so much smaller in comparison with the unweatherstripped window that a small variation in this crack will cause a measurable variation in the leakage. The curves for the weatherstripped windows show a corresponding increase with size of crack.

Tables 1 and 2 contain data taken from the curves Figs. 6 to 11 or resulting therefrom. Table 1 gives the leakage in cubic feet per minute for the whole window and per linear foot to crack, for wind velocities of 14.4 and 24.9 miles per hour. It is of interest to note that for a plain window with crack varying from $\frac{1}{16}$ to $\frac{1}{4}$ in. the leakage is 46 cu. ft. per min., while for the two types of weatherstripping tested it varies from 9 to 18 and 7 to 10 cu. ft. per min. respectively. The heat loss is given for two temperature differences. The heat loss for any temperature difference varies directly as the leakage. The radiation required to supply this heat loss is given for the higher temperature difference only, since it must be supplied for the maximum condition. With a 14.4 mile wind based upon the above temperature difference the unweatherstripped windows with cracks varying from $\frac{1}{16}$ to $\frac{1}{4}$ in. required 14.6 sq. ft. of radiation, while the same windows with the two types of weatherstripping require only from 2.8 to 5.7 and 2.2 and 3.2 sq. ft. respectively. Basing the cost of radiation on \$2.00 per sq. ft. installed, the two types of weatherstripping will show a resulting decrease in first cost of radiation of about \$18.00 and \$23.00 per window respectively. The further saving in coal per year based upon a seven month heating season with an average temperature difference of 35 deg. is also given.

Table 2 gives the "elsewhere," wall, and frame leakage, and also the leakage through with the window with and without weatherstripping for various wind velocities.

Perhaps the most surprising fact brought out by this table, if not by the whole investigation, is the leakage per square foot of wall. With a 15 mile wind each

TABLE 1. DATA ON TESTS OF WINDOWS UNDER VARIOUS CONDITIONS

	Leakage C.F.M. Crack Perimeter 18" 4"		B.t.u. per Hour		Radiation on Sq. Ft. 240 B.t.u. per Sq. Ft. per Hour		Lb. of Coal Based on 13,000 B.t.u. and 50% Efficiency at Radiator 35°-70°Fahr. for Seven Months	
	0-70°Fahr.		0-70°Fahr.		0-70°Fahr.		0-70°Fahr.	
	For total window	Per ft. crack	For total window	Per ft. crack	For total window	Per ft. crack	For total window	Per ft. crack
Wind velocity, M.P.H....	14.4	24.9	14.4	24.9	14.4	24.9	14.4	24.9
Plain window—very tight	19.0	40.0	1.04	2.18	1445	3040	78.7	166.0
Plain window—tight—un- locked.....	22.0	46.0	1.20	2.53	1673	3500	91.3	192.5
Plain window—1/16" to 1/8" crack—unlocked...	46.0	75.0	2.53	4.08	3500	5700	192.5	310.0
1/16" Crack...	9.0	20.0	0.49	1.09	685	1322	37.3	83.0
1/8" Crack...	11.0	22.5	0.60	1.25	837	1710	45.6	95.1
3/16" Crack...	15.0	30.0	0.82	1.63	1141	2282	62.2	124.4
1/4" Crack...	18.0	36.0	0.98	1.96	1370	2740	74.7	149.4
1/2" Crack...	7.0	16.0	0.38	0.87	533	1218	29.0	66.3
1/8" Crack...	8.5	18.0	0.46	0.98	647	1370	35.3	74.7
1/4" Crack...	9.0	19.5	0.49	1.06	685	1483	37.3	80.9
1/2" Crack...	10.0	20.5	0.55	1.12	761	1560	41.4	85.0

TABLE 2. LEAKAGE IN C.F.M.

Wind Velocity M.P.H.	Frame 17 Ft. Perimeter	Wall per Sq. Ft.	Else- where		Window with 1/16 In. Crack		Perimeter 18 Ft. 4 In.	
			For total window	Per ft. crack	For total window	Per ft. crack	For total window	Per ft. crack
5	0.50	0.0111	0.7	0.818	1.3	0.071	1.0	0.0546
7.5	2.3	0.0289	1.8	1.31	3.0	0.164	2.2	0.1200
10	4.0	0.0512	2.8	1.77	5.0	0.273	4.0	0.2180
15	8.0	0.1110	5.2	2.59	9.7	0.529	7.8	0.4260
20	11.0	0.1780	7.6	3.36	14.8	0.808	11.8	0.6440
30	19.0	0.3330	13.6	4.86	25.5	1.390	20.0	1.0920
40	25.5	0.5250	20.0	6.44	37.0	2.020	30.0	1.6380
50	31.5	0.7150	26.0	8.13	48.5	2.650	41.0	2.2400

square foot of the 13 in. wall, plastered on the inside, allowed the passage of 0.111 cu. ft. of air per min., while the leakage through the window and frame for the same wind velocity was 47.5, 9.7, and 7.8 cu. ft. per min. for the plain window and two types of weatherstripping respectively. The area of the window and frame is 16.25 sq. ft. giving a leakage of 2.82, 0.597 and 0.48 cu. ft. per min. per sq. ft. of window without and with the two types of weatherstripping. Based upon these figures the leakage through the window and frame varies from 4 to 28 times that through the same area of wall. When we take into consideration the usual greater area of wall to window, it is evident that the leakage into a room is usually greater through the wall than through the window if weatherstripped, and not many times less if not weatherstripped. It is of interest to compare the heat loss through windows and walls by transmission and by leakage. The leakage for the plain window and

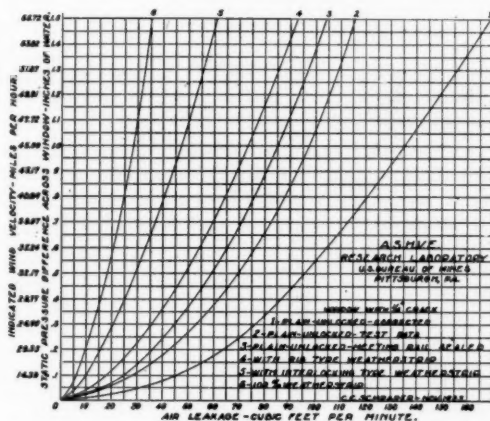


FIG. 11. RESULTS OF TESTS ON WINDOW WITH $\frac{1}{16}$ " CRACK AROUND PERIMETER

with two types of weatherstripping all for $\frac{1}{16}$ in. crack and a 15 mile wind is 47.5, 9.7, and 7.8 cu. ft. per min. respectively, representing a heat loss of 2580, 527, and 423 B.t.u. per hr. respectively for a 50 deg. temperature difference. A leakage of 0.111 cu. ft. per min. per sq. ft. of wall represents a heat loss of 6.03 B.t.u. per hr. for a 50 deg. temperature difference. Taking the transmission through the wall as 0.28 B.t.u. per hr. per sq. ft. per degree temperature difference, this loss is 14 B.t.u. per hr. for the same temperature difference. The heat loss as thus indicated by infiltration is 43 per cent as great as the heat loss by transmission as indicated by the constant used.

The values given in the table are from the tests and are probably somewhat higher than those actually found in practice. They represent the leakage when the pressure drop through the window is a certain value which represents a definite wind velocity at right angles to the window. If the wind strikes the window at an oblique angle the component of the velocity at right angles to the window must be considered. Pressure difference between the outside and the inside surfaces of the

window for an actual wind will be slightly less for a given velocity because of a building up of pressure within the room before the air leaks out the opposite side of the building. Attention is called to the fact that air leaks in on the windward side of the building and out on the leeward side and, since wind will blow from various directions at different times, heating for any room having only one exposure must be based on the maximum loss. The heating plant, however, need not be figured on the sum of all maximum leakages but in general only half of the total. However, the tables give accurate comparative figures which are probably not much too high for actual practice. In order to apply these values, a further study of the overall results as found in practice should be made, and the figures modified, if necessary, to fit practical conditions.

DISCUSSION

W. H. CARRIER: About 15 years ago, I believe, we had a paper presented before this Society on window leakage. It would be very interesting to compare the work that was done at that time with what we have tonight.

E. S. HALLETT: I have been talking tight windows for several years, because they affected my work so much. When we went to recirculate all of the air, it became necessary to have tight windows in order to obtain good results. The fact that caulking frames and the improved window strip has done so much good is amazing.

No. 687

THE PRODUCTION AND MEASUREMENT OF AIR DUSTINESS

By MARGARET INGELS,¹ PITTSBURGH, PA.

MEMBER

AIR with various degrees of dustiness is needed if an instrument for measuring the amounts of dust is to be tested over a wide range. It is unsatisfactory to depend on naturally dusty air because its degree of dustiness is constantly changing.

Dust Liberator

To introduce dust in the air is a simple problem. To produce a uniformly dusty air for a period of several hours is a problem whose difficulty can be realized only by those scientists who have worked extensively with dust.

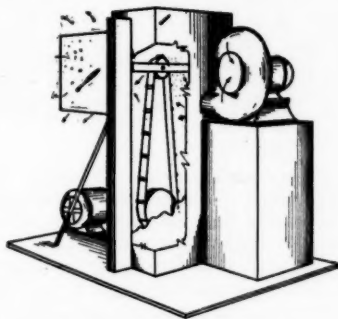


FIG. 1. DUST LIBERATOR

An apparatus has been developed in this laboratory for adding dust to room air. A diagrammatical drawing of this apparatus is shown in Fig. 1. A motor-driven worm and gear turns a pulley buried in a supply of dust, in a galvanized iron box. Near the top of the box, and above the dust level is a second pulley. On these two

¹ Research Head, A.S.H.&V.E. Laboratory.
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pulleys rides a belt carrying small buckets. The buckets pick up a load of dust, and empty it, refill and empty again continuously. This keeps a large amount of dust floating in the air in the box. On one side of the box, air is blown in by a small centrifugal fan, which keeps a constant pressure in the box. On the opposite side of the box is a small outlet through which the air and floating dust flow. The constant pressure keeps the air flow constant; the amount of floating dust is constant, therefore the dust liberated is constant. A screen is placed about 10 inches from the outlet to remove by impact large particles of dust that are carried only by the high air velocities. There is nothing about the apparatus to clog up with dust. A special arrangement connected to a pulley prevents the dust in the reservoirs from banking up. The continuous pressure in the box prevents puffs of dust. The amounts of dust are varied by changing the pressure.

There is no way of determining the amount of dust liberated, but the dust that remains in the air is all that will be of interest.

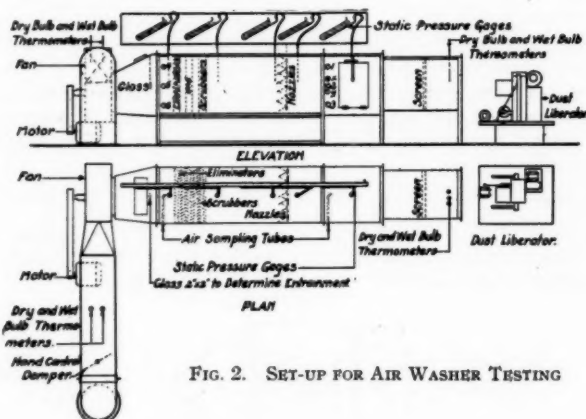


FIG. 2. SET-UP FOR AIR WASHER TESTING

Dusts

In the following tests three dusts are used. One called "Room Dust" is the dust found in usual Pittsburgh air. The others are two sizes of powdered coal dust. It is possible to get coal dust in large quantities, a small deposit can be seen on white filter paper, and the U. S. Bureau of Mines equipment can be used to size it. One coal dust is that which was passed through a 200-mesh sieve and remained on top of a 300-mesh sieve. Its particles are from 70 to 140 microns in diameter. The other coal dust is that which was passed through a 300-mesh sieve. Its particles are 70 microns and less in diameter. Under the best conditions the smallest particle a human eye can see is from 30 to 50 microns in diameter.

Air Circulation System

The application of the Anderson and Armspach Dust Determinator for measuring the efficiency of air cleaning equipment is to be studied in this laboratory. For this work an air circulating system has been installed. A drawing of this installation is shown in Fig. 2. The dust liberator is placed at the air inlet and samples taken in the duct at points 1-2-3-4-5-6, *x* and *y*. The air is drawn

through a typical air washer into a fan and blown through a duct into the tunnel under the floor. The air quantity is varied by the use of the hand-controlled damper in the outlet duct.

Dust Chart

Six laboratory dust determinators, shown in Fig. 3, are set up for this work with dust. These six determinators should check each other if all samples are taken from the same air. Tests were run as follows to see how the determinators would check. When room dust is used, the dust determinators are placed as close together as possible and air from the room is drawn directly into them. When the coal dusts are used the air is drawn from the intake of the washer, into a distributing box made of galvanized iron. All six determinators draw air from this distributing box and should get air of equal dustiness.

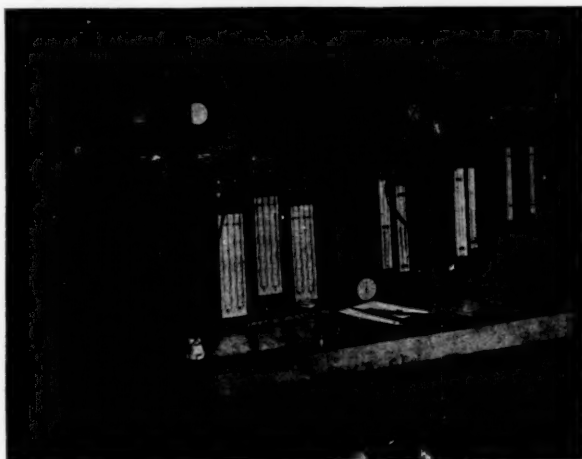


FIG. 3. GROUP OF SIX DUST DETERMINATORS IN LABORATORY

Figs. 4, 5 and 6 show the filters from room dust—small size coal dust and large size coal dust, respectively. Three tests were taken for each dust, and six determinations made in each test. The shades of the filters for each test indicate that air of equal dustiness is sampled by all of the determinators.

The curves in Fig. 7 show the data for these tests. All tests are one-half hour in length and pressure drops in inches of water across the filter are read every five minutes. The filter medium is A. D. Little, 7 cm., chemical filter paper and the air flow is kept constant at 0.4 cu. ft. per min.

It will be seen from curves in Fig. 7 that the pressure drops across clean filters (that is zero time readings) vary from 3.55 in. to 6.2 in.; also that for any one test the higher the initial pressure drop the greater the rate of increase in pressure difference for the same dustiness.

It is not possible to get filters which will always have the same initial resistance for an equal rate of air flow. It is necessary to determine a relation between the rate of increase in resistance for a dust deposit and the initial resistance of the paper.

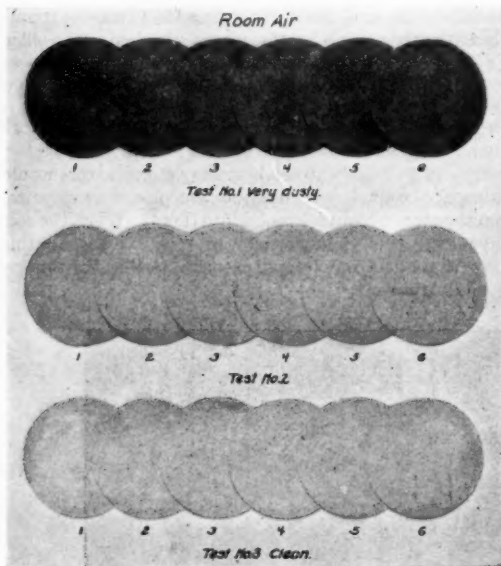
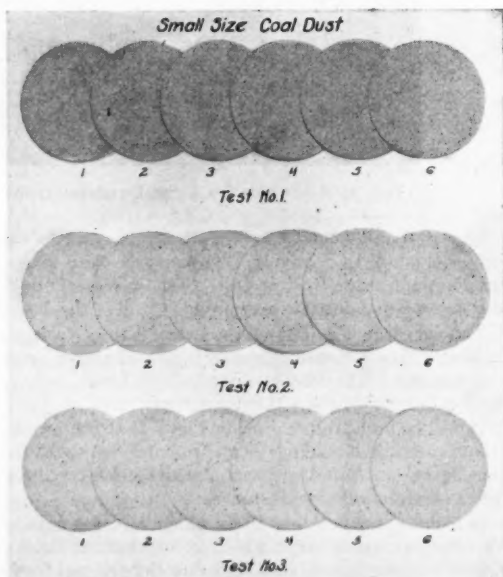


FIG. 4. RESULTS OF THREE TESTS OF SIX DUST DETERMINATORS OPERATED SIMULTANEOUSLY—ROOM AIR

FIG. 5. RESULTS OF TESTS OF SMALL SIZE COAL DUST

Coal dust passed through 300 mesh sieve and added to clean room air—maximum size = 70 microns, minimum size = 10 microns



The pressure drops for different rates of air flow are determined for several A. D. Little chemical filter papers. The actual test data and four estimated curves are shown in Fig. 8. At low pressures these curves approach straight lines, but as the pressures increase there is less increase in the volume of air. This is characteristic of orifice curves, therefore the filters will be considered as orifices and assumptions made accordingly.

An orifice is usually used to measure the quantity of air. The pressure, area and weight of air are determined from tests. The coefficient of discharge for thin plate

orifices and pressures not above six inches is approximately 0.6. The equation is used to solve for Q .

The filter considered as an orifice is more complicated. The air quantity remains constant, the area changes, the coefficient of discharge changes and the pressure drop changes. A clean paper has a fixed orifice area as long as it remains clean, but when dust is deposited, the area of the orifice becomes smaller, and as the deposit increases the area of the orifice approaches zero. At the same time the pressure is increasing. As the area of the orifice becomes smaller, the coefficient of discharge becomes larger, and approaches 1 as a limit.

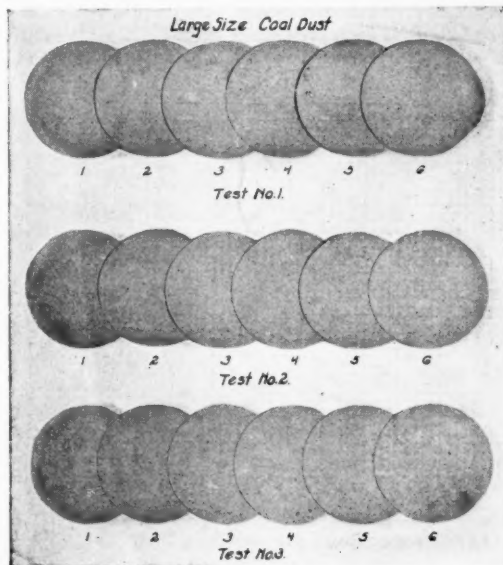


FIG. 6. THREE TESTS OF LARGE SIZE COAL DUST

Coal dust that passed through 200 mesh screen and stayed on top of 300 mesh added to clean room air—maximum size = 140 microns, Minimum size = 70 microns

In the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS' GUIDE, 1922, p. 170 the equation for the discharge of an orifice is:

$$Q = 1096.5 k A \sqrt{\frac{p}{W}}$$

Q = Cu. ft. per min.

k = coefficient of discharge.

A = area of orifice in sq. ft.

p = pressure in inches of water.

W = Weight of air in lb. per cubic feet = 0.07488 standard condition.

The pressure and cubic feet of air flow for each filter are read from curves in Fig. 8 and are tabulated as shown in Table 1. Column 1 shows the name of the filter, Column 2 gives the pressure and Column 3 gives cubic feet of air. Using the orifice formulae in the form of:

$$kA = \frac{\text{c.f.m.} \times 0.0002495}{\sqrt{p}}$$

and solving for kA the figures for Column 4 are obtained. An arbitrary value of k must be assumed, and its variation for each filter, air quantity and pressure figured. The greater the pressure the greater the coefficient of discharge. For filter C and 14 in. pressure, k is assumed 0.90. If this value is either too high or too low the other values of k will also be too high or too low—but the ratios of the values will not be changed. With $k = 0.90$ for filter C and 14 in. pressure, and kA 0.0000423, A will equal 0.0000496 sq. ft. This area remains constant for there is no dust deposit, so the value of k can be found for each pressure of filter C . k equals 0.37 when $p = 2$ in. The area of each filter is figured by assuming $k = 0.37$ for 2 in.

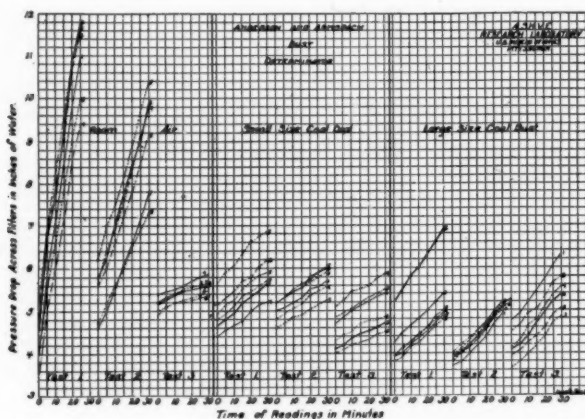


FIG. 7. RESULTS OF TESTS OF ROOM AIR AND COAL DUST

pressure. As the area remains constant for each filter the value of k can be figured for each pressure. These values are given in Column 5. In Column 7 is given the pressure for 0.4 cu. ft. per min. for each filter read from curves in Fig. 8.

In Fig. 9 are shown curves with the values of k plotted against cubic feet of air flow, for each filter. The Anderson and Armspach Dust Determinator uses 0.4 cu. ft. per min. through the filters so the values of k are needed for this air quantity only. With the values read from curves in Fig. 9, the relations of k to the orifice area for 0.4 cu. ft. per min. and the relations of k to the pressure difference for 0.4 cu. ft. per min. are shown by curves in Fig. 10. Using these curves a relation of pressure difference to area of orifice is determined and shown in Fig. 11. With this curve a theory will be set up and actual tests made to see if the theory works out in practice.

A filter paper consists of an infinite number of small holes through which the sample of air passes. These holes will be considered as one opening whose area is equal to the total area of the small holes. This total area will act according to laws of an orifice, and will be referred to as the free area of the paper.

All filters requiring the same pressure to produce 0.4 cu. ft. per min. air flow will

have equal free areas. If a filter has such an area as to require x inches of pressure to give 0.4 cu. ft. per min. flow, and dust is deposited on the filter until a pressure of $2x$ inches is needed, then the area has been decreased. Equal dust deposits will decrease the free area of the filter by a constant area, regardless of the initial free area of the filter. Consider a filter which has a comparatively large area of x square feet and dust is deposited on the filter to decrease the area by a , the new area of the paper is $x - a$. A second filter whose area is $x/3$ collects the same amount of dust and its area becomes $x/3 - a$. The decrease in area of each filter is equal, but the increases in pressure do not correspond as the coefficient of discharge— k —of the orifice formulae does not remain a constant. k varies and becomes larger as the area becomes smaller. The differences in pressure due to constant decreases in area vary and become greater as initial areas become smaller.

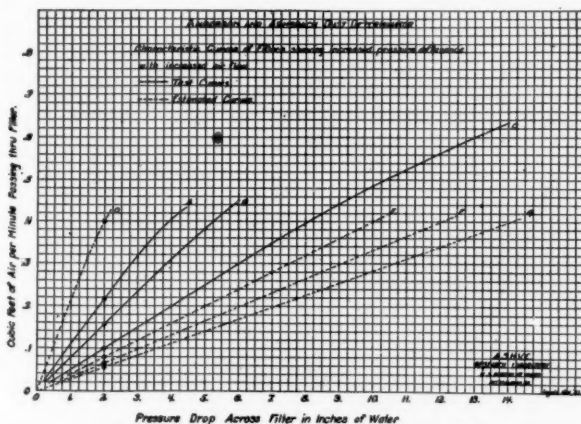


FIG. 8. CHARACTERISTIC CURVES OF FILTERS SHOWING INCREASED PRESSURE DIFFERENCE WITH INCREASED AIR FLOW

The free area of a filter having 2 in. pressure drop is equal to 0.0001912 sq. ft. The free area of a filter having 3 in. pressure drop is equal to 0.0001390 sq. ft. These areas are read from the curve in Fig. 11. If a filter has a pressure of 2 in. and collects dust until it has a pressure of 3 in., the area has been decreased 0.0000512 sq. ft., that is the area of the dust deposit is equal to 0.0000512 sq. ft. By using each pressure and determining the decrease in area for each other pressure, figures for the initial resistance lines of the dust chart are obtained. This chart is shown in Fig. 12. If k were a constant the decreases in area would be read along horizontal lines. Corrections must be made for the variations of k . The coefficient of discharge is the per cent of orifice area that is effective. If comparisons are to be made of filters with unequal areas, a relation of their effective areas must be found. Curves drawn showing the variations of k will also give the relations of effective areas. Comparisons of filters with unequal areas can be made along these curves. They are called lines of equal dustiness on the dust chart.

If a filter has an initial pressure of 2 in. and a dust deposit causes the pressure to increase to 3 in. for 0.4 c.f.m. air flow, the amount of dust deposited is found as follows:

On the zero line of the decrease in area scale, find the point where the initial pressure is 2 in. Follow this line until it crosses the 3 in. pressure line. Read the decrease in area caused by this dust; this is equal to 0.00052 sq. ft. It will be seen from the equal dustiness line that for this same dust deposit a filter with an initial pressure drop of 7 in. would have a final pressure of 10.2 in. To make any readings of dustiness find the initial pressure line for the filter used and follow this line until it crosses the average pressure difference line. From this point of intersection follow along the equal dustiness line to the 2 in. initial pressure line as a base and read the decrease in area due to the dust.

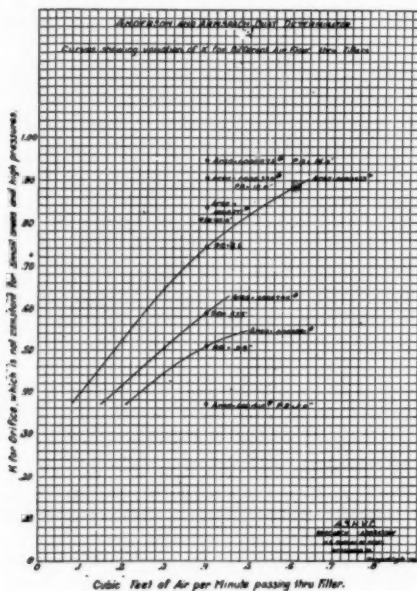
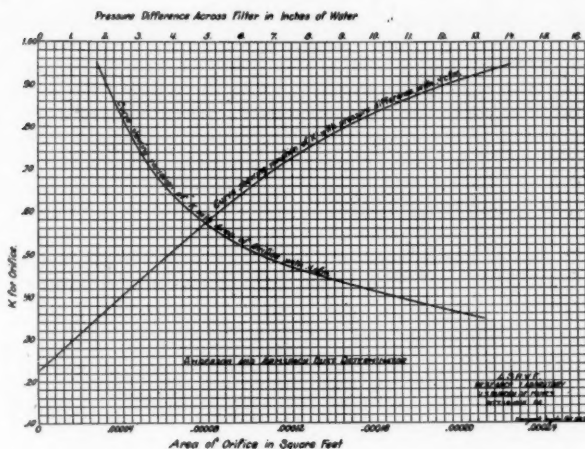


FIG. 9. CURVES SHOWING VARIATION OF k FOR DIFFERENT AIR FLOW THROUGH METER

Ten tests on room air, three tests on air containing small size coal dust, and four tests on air containing large size coal dust are made to learn if the dust chart will check.

Six simultaneous determinations of dust are made in each test. The dust chart is used to correct the determinations where the filters have various initial resistances to a 2 in. initial resistance base. In every test the six determinations checked each other with a maximum variation of 10 per cent from the average, and a minimum variation of 5 per cent from the average. In Table 2 is given the data of nine of the tests made to check the dust chart.

This is better than any dust tests have ever checked regardless of the method used to measure the dust. The possible error in test data is greater than the


 FIG. 10. CURVES SHOWING VARIATION OF k

variation in results taken from the chart. At the present time this dust chart will be used in connection with the Anderson and Armspach Dust Determinator.

When further work is done the chart will be divided into zones showing clean air, and the varying degrees of dustiness.

Sampling Moving Air

To determine the dustiness of moving air introduces various complications in taking the sample. The sample must be drawn out and preferably at the same

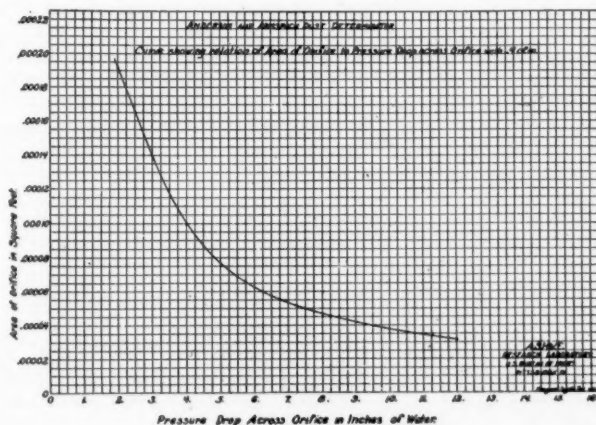


FIG. 11. CURVE SHOWING RELATION OF AREA OF ORIFICE TO PRESSURE DROP ACROSS ORIFICE WITH 0.4 C.F.M.

TABLE 1. PRESSURE AND CUBIC FEET OF AIR FLOW FOR EACH FILTER

1	2	3	4	5	6	7
Filter	ϕ	C. F. M.	$k \times a$	$k =$	Area	Pressure at 0.4 cu. ft. min.
A	2"	0.208	0.0000367	0.37	0.0000991	3.9
	3"	0.314	452	0.456	991	
	4"	0.408	510	0.515	991	
	5"	0.485	540	0.545	991	
B	2"	0.153	0.00002705	0.37	0.0000744	5.25
	3"	0.227	3270	0.44		
	4"	0.307	3820	0.514		
	5"	0.382	4270	0.575		
	6"	0.451	457	0.615		
	2"	0.098	0.00001733	0.37		
C	3"	0.148	2130	0.454		8.1
	4"	0.198	2465	0.524		
	5"	0.248	2770	0.59		
	6"	0.298	3040	0.647		
	7"	0.347	3270	0.696		
	8"	0.394	3480	0.741		
	9"	0.437	364	0.775		
	10"	0.482	380	0.81		
	11"	0.523	393	0.836		
	12"	0.552	399	0.85		
D	13"	0.60	415	0.884		Started here
	14"	0.634	423	0.90*	0.0000469	
	2"	0.4	0.0000706	0.37	0.000191	2
E	2"	0.057	0.00001008	0.37	0.0000272	14
	14"	0.4	267	0.945		
F	2"	0.067	0.00001183	0.37	0.000032	12
	12"	0.4	289	0.905		
	2"	0.079	0.00001395	0.37	0.0000377	
	10"	0.4	316	0.837		

velocity as that of the moving air. The sampling tube would have to be of various sizes to care for all velocities, which is not practical. If different inlets were used on the tube there would be a change of velocity inside the tube. This change in velocity would possibly change the degree of dustiness at the dust determinator. It is not possible to change the rate at which the sample is taken for each velocity because dust measuring apparatuses are limited in their air capacities.

Sampling tubes for the work with the Anderson and Armspach Dust Determinator are made of $\frac{5}{8}$ in. brass tubing, Fig. 13. The ends of the tubes are bent through 90 deg. with a 3 in. radius.

The dust determinator will sample the air at the rate of 0.4 cu. ft. per min., this means a linear velocity in the sampling tube of 190 ft. per min.

The control valve on the dust determinator was adjusted to handle 0.4 cu. ft. per min. when sampling still air. The air was given a velocity of 600 ft. per min. and the sampling tube pointed into the air flow to see if the total pressure of this air would act with vacuum caused by the dust determinator and increase the amount of air through the determinator. The air quantity did not change. The filter medium causes such a large part of the resistance through the sampling system that a change of resistance in any other part of the system does not change the rate of sampling.

When the velocity of the moving air is less than the velocity into the sampling tube the following may be expected to happen:

When the tube is pointed into the air flow the sample may draw more air for the amount of dust taken due to the air being lighter than the dust. If the air blows across the end of the tube there will be the same effect, and to a greater extreme. The dust measurements will be too low.

When the velocity of moving air is greater than the velocity into the sampling tube the effect is different. When the tube is pointing into the flow the dust will continue straight and go into the tube, the air will tend to carry on around the tube by the higher velocity. This means the dust measurement will be too high. When the air blows across the end of the tube, the effect will be to get more air and less dust; the dust measurement will be too low.

The effects of the direction of pointing the sampling tube and the effects of different velocities of the moving air are determined by test.

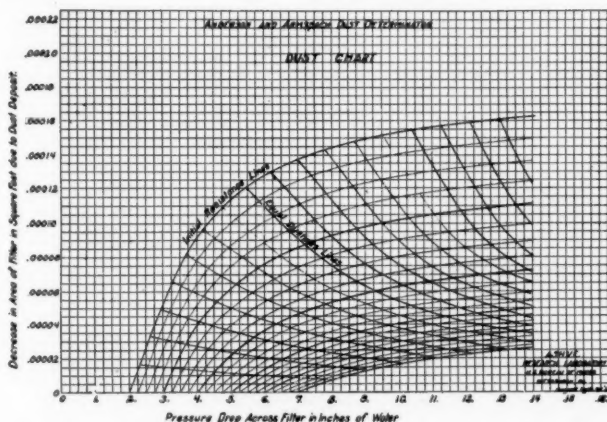


FIG. 12. THE DUST CHART

Sampling tubes are placed in the duct according to the drawing in Fig. 13. The top, middle and bottom tubes are kept pointing into the flow during all the tests.

Samples taken through these tubes will show the degree of dustiness across the area of the duct. The directions of the "upper" tube and "lower" tube are changed for each test. In one test they are pointing into the flow, in the second test they are at 90 deg. to the direction of air flow, in the third test they are 45 deg. to the direction of air flow. For each position of the variable tubes, two velocities are used across the tubes, for air containing large size coal dust and for air containing small size coal dust.

The data obtained in these tests, and corrected to the base of 2 in. initial resistance by using the dust chart, are shown by the curves in Fig. 14. The upper and lower tubes are marked 90 deg. when pointing at right angles to the air flow, and 45 deg. when making an angle of 45 deg. with the air flow. Table 3 is made from the curves showing the degrees of dustiness at the various points of sampling.

In the first column of Table 3 is given the size dust added to room air, the second column gives the velocity of the air in the duct.

Columns 3-6, and 9 give the amounts of dust found in the sample, when the sam-

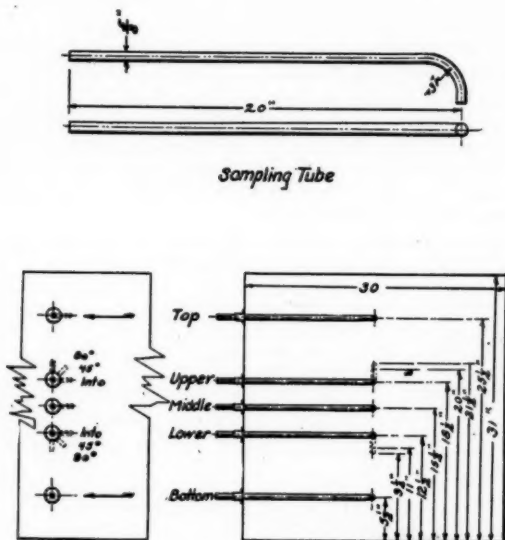


FIG. 13. SAMPLING TUBES AND ARRANGEMENT FOR TEST

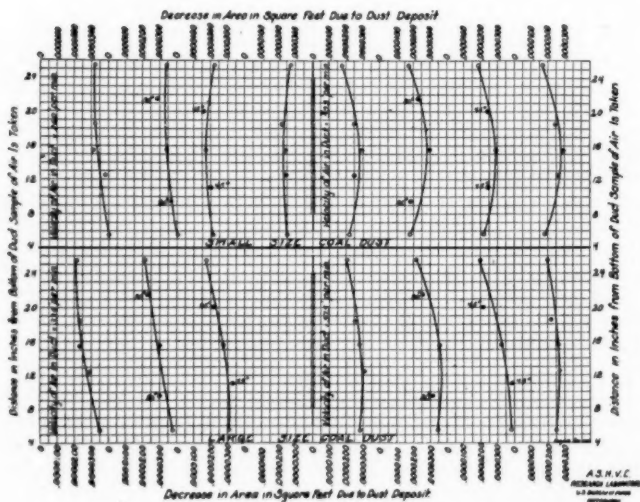


FIG. 14. CURVES SHOWING DEGREE OF DUSTINESS AT POINTS OF SAMPLING

pling tubes are into, at 90 deg. and at 45 deg. to the direction of air flow. In Columns 4, 7 and 10 are given the amounts that would make these points fall on the line, and are used as the correct readings. The per cents the data readings differ from the correct readings are given in Columns 5-8, and 11. These per cents are averaged and it will be noted that the tubes pointing into the flow average very closely, again showing the accuracy of the dust chart. The tubes at 45 deg. get as much dust as those at 0 deg. at low velocities but less at higher velocities. The tubes at 90 deg. have a lower dust determination even at low velocities, which is still lower at high velocities.

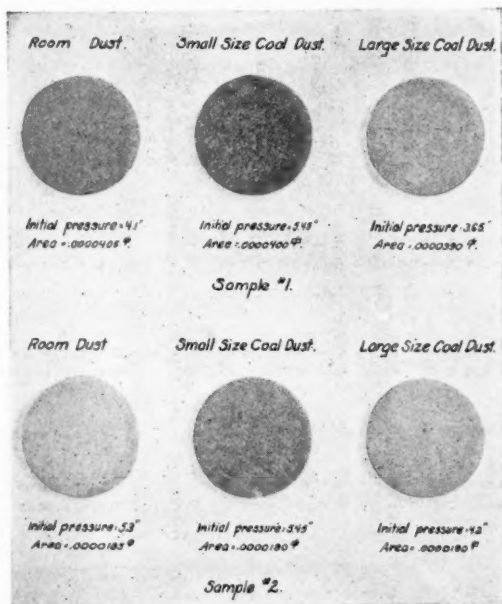


FIG. 15. TWO DEGREES OF DUSTINESS RESULTING FROM TESTS

These tests show what was expected would happen for velocities over the tubes greater than the velocity in the tubes. The "high readings of dust" for the air when the tube is pointing into the air flow and the "low readings of dust" for the air when the tubes are 90 deg. with the air flow are not far apart. The method of using the tubes pointing into the flow for taking samples is permissible and the error will be small unless the velocity in the duct is very high, higher than ever need be sampled.

The color of a filter on which dust has been deposited is dependent on the amount, the kind and the size of the dust. In Fig. 15 are shown two degrees of dustiness for the three dusts used throughout this work. The first is a deposit which will decrease the areas 0.000400 sq. ft., the second will decrease the areas 0.000185 sq. ft. (see Table 4). It will be seen from the filters shown that the colors are not

TABLE 2. (Continued)

Time	1	2	3	4	5	6	Remarks
0	4.65	5.10	4.8	4.4	4.9	5.65	70 Micron Dust from Equalizer Box
5	4.8	5.3	4.95	4.55	5.15	5.9	
10	4.95	5.45	5.1	4.65	5.25	6.05	
15	5.2	5.7	5.35	4.75	5.5	6.35	
20	5.3	5.8	5.45	5.0	5.7	6.6	
25	5.45	6.05	5.6	5.15	5.85	6.75	
30	5.65	6.2	5.7	5.2	5.9	6.85	
Avg.	5.14	5.65	5.28	4.82	5.46	6.30	Area from Chart Avg. = 0.0000151. Maximum off = 7%
0	5.0	5.2	4.6	4.8	5.2	5.2	70 Micron Dust from Equalizer Box
5	5.05	5.3	4.7	4.95	5.3	5.3	
10	5.2	5.45	4.85	5.05	5.5	5.45	
15	5.3	5.55	4.9	5.15	5.65	5.55	
20	5.5	5.8	5.05	5.35	5.75	5.70	
25	5.55	5.9	5.15	5.45	5.9	5.80	
30	5.7	6.0	5.25	5.55	6.05	5.9	
Avg.	5.33	5.60	4.93	5.19	5.62	5.56	Area on Chart Avg. = 0.0000109. Maximum off = 10%
0	4.7	4.1	4.85	5.1	4.00	4.3	70 Micron Dust from Equalizer Box
5	4.85	4.2	4.95	5.25	4.1	4.45	
10	5.0	4.35	5.05	5.45	4.2	4.5	
15	5.1	4.45	5.2	5.50	4.25	4.7	
20	5.2	4.5	5.3	5.60	4.3	4.75	
25	5.3	4.6	5.4	5.75	4.45	4.75	
30	5.4	4.7	5.5	5.85	4.5	4.85	
Avg.	5.09	4.41	5.19	5.51	4.26	4.61	Area on Chart Avg. = 0.0000117. Maximum off = 8%
	0.0000120	0.0000120	0.0000120	0.0000120	0.0000110	0.0000115	

TABLE 2. (Concluded)

Time	1	2	3	4	5	6	Remarks
0	4.25	5.20	3.8	3.9	3.9	3.9	140-70 Micron Dust from Equalizer Box
5	4.45	5.5	3.9	4.1	4.0	4.15	
10	4.65	5.8	4.15	4.25	4.2	4.25	
15	4.8	6.0	4.25	4.4	4.35	4.45	
20	5.0	6.3	4.45	4.6	4.5	4.6	
25	5.2	6.65	4.65	4.8	4.75	4.85	
30	5.4	6.9	4.8	4.95	4.9	5.0	
Avg.	4.82	6.05	4.29	4.43	4.37	4.46	Area from Chart Avg. = 0.0000187. Maximum off = 9%
	0.0000180	0.0000200	0.0000185	0.0000195	0.0000170	0.0000195	140-70 Micron Dust from Equalizer Box
0	3.7	3.9	3.95	3.9	3.8	4.0	
5	3.8	4.0	4.1	4.05	4.0	4.2	
10	4.0	4.2	4.3	4.2	4.15	4.35	
15	4.2	4.4	4.5	4.4	4.3	4.5	
20	4.4	4.6	4.7	4.7	4.55	4.7	
25	4.7	4.9	4.95	5.0	4.85	4.95	
30	4.9	5.1	5.2	5.2	5.0	5.1	
Avg.	4.24	4.44	4.53	4.49	4.38	4.54	Area from Chart Avg. = 0.0000198. Maximum off = 9%
	0.0000180	0.0000195	0.0000200	0.0000210	0.0000210	0.0000195	140-70 Micron Dust from Equalizer Box
0	4.75	4.05	3.65	4.1	3.95	4.50	
5	4.9	4.2	3.80	4.3	4.05	4.65	
10	5.2	4.4	4.0	4.65	4.25	4.9	
15	5.45	4.7	4.2	4.75	4.4	5.05	
20	5.75	4.9	4.4	5.1	4.6	5.3	
25	6.05	5.2	4.7	5.35	4.9	5.65	
30	6.3	5.55	4.85	5.55	5.05	5.8	
Avg.	5.49	4.69	4.26	4.83	4.47	5.12	Area from Chart Avg. = 0.0000208. Maximum % = 10%
	0.0000200	0.0000210	0.0000230	0.0000230	0.0000190	0.0000190	

TABLE 3.

2	3 All Tubes Into Should be from curves	4 % of Cor- rect reading	5 Data	6 Tubes 90° Should be from curves	7 % of Cor- rect reading	8 Data	9 Tubes 45° Should be from curves	10 % of Cor- rect reading	11 Avg.
0.0000230	0.0000230	97.7	0.0000230	0.0000242	95.0	0.0000220	0.0000237	92.8	Avg. = 100.4%
230	250	92.0	300	330	91.0	330	305	108.0	
290	270	107.0							
300	270	110.0							
		Avg. = 101.5%			Avg. = 93.0%				
0.0000260	0.0000270	96.0	0.0000215	0.0000270	79.6	0.0000215	0.0000255	84.3	Avg. = 95.4%
270	280	96.5	210	250	84.0	290	270	107.5	
310	290	107.0							
220	235	93.5							
265	250	106.0							
		Avg. = 99.8%			Avg. = 81.9%				
0.0000310	0.0000335	92.5	0.0000285	0.0000330	86.3	0.0000260	0.0000280	93.0	Avg. = 100%
385	350	110.	370	370	100.0	200	185	108.0	
225	245	91.6							
265	235	113.0							
		Avg. = 101.7%			Avg. = 93.1%				
0.0000295	0.0000265	110.0	0.0000230	0.0000250	92.0	0.0000240	0.0000255	94.0	Avg. = 94%
250	270	92.5	185	250	74.0	245	260	94.0	
240	260	92.5							
290	265	105.5							
		Avg. = 100.1%			Avg. = 84%				

140 Micron Dust

70 Micron Dust

equal in shade, and that they are not consistent for the tests where coal dusts of two sizes are used.

Equal volume of dust deposited under the same conditions will cover equal areas regardless of the weight, size, kind or color of dust. Area is the absolute scale which will be used to measure dustiness with the Anderson and Armspach Dust Determinator.

TABLE 4. SHOWING DATA FOR TESTS OF EQUAL DUSTINESS WITH DIFFERENT KINDS OF DUST

Time	Room Dust	70 Micron Dust	140 Micron Dust
0	4.1	5.45	3.65
5	4.5	5.95	4.0
10	4.95	6.65	4.35
15	5.5	7.4	4.9
20	6.0	8.0	5.2
25	6.5	8.8	5.6
30	6	9.4	5.8
Avg.	5.51	7.36	4.79
Area	0.0000405	0.0000400	0.0000390
0	5.3	5.45	4.2
5	5.5	5.7	4.35
10	5.8	6.0	4.55
15	6.1	6.3	4.75
20	6.35	6.55	4.9
25	6.45	6.9	5.3
30	6.6	7.2	5.5
Avg.	6.02	6.3	4.79
Area	0.0000185	0.0000190	0.0000190

DISCUSSION

E. VERNON HILL: If I understand the Anderson and Armspach dust determinator correctly, it was originally designed for measuring air dustiness by means of a drop in pressure between the sides of a filter, the filter collecting the dust, increase in resistance showing a corresponding increase in the measurements. It develops, if I have followed this closely, that due to differences in the filter material, possibly due to its hygroscopic properties, etc., this is impractical and now Miss Ingels proposes to substitute a volumetric determination instead of a determination by means of a drop in resistance. In other words, the Anderson and Armspach dust determinator does not work out satisfactorily on the original basis. It seems to me this dust determination problem is getting quite intricate and I am going to suggest (I conceived this in a spirit of fun but perhaps there might be something to it) that we have a dust determination marathon at the summer meeting. Let's take this apparatus which appears to be admirable for distributing dust in a room and invite all the various persons who would be interested in developing dust determination devices to get in the room and make determinations and check up results. Of course, it would not be competitive; it would be comparative and prove to be very interesting.

MISS INGELS: Last summer six different kinds of dust measuring apparatus were tested simultaneously by the U. S. Bureau of Mines. They were the Sugar Tubes, Palmer, the Konimeter, the new Impinger made by Doctor Greenburg and Mr. George Smith, and the A-A Dust Determinator. The A-A instrument was in its very first stages of development so a real comparison of it with the other instruments can only be made approximately with these tests.

No. 688

CRITICAL VELOCITY OF STEAM AND CONDENSATE MIXTURES IN HORIZONTAL, VERTICAL, AND INCLINED PIPES

By F. C. HOUGHTEN¹ (MEMBER), LOUIS EBIN,²
AND R. L. LINCOLN³ (NON-MEMBERS).

PITTSBURGH, PA.

A REPORT of the flow of steam in vertical pipes with counterflowing condensate was presented at the Annual Meeting of the Society, held at Washington, D. C., in January, 1923, and was published in the March, 1923, issue of the Society's JOURNAL. This report was devoted to the factors affecting the flow of steam, entirely in vertical pipes. Some of the important deductions brought out in that report are:

1. The critical velocity for vertical risers varies from 22 to 30 ft. per second.
2. The critical velocity of a pipe is determined by the smallest area of that pipe.
3. The critical velocity of a pipe not reamed is decreased by approximately 10 per cent for a squared entrance and by as high as 30 per cent if a burr due to using a wheel cutter is left unreamed.
4. Unions and other fittings used in a riser should have full size openings.

Further work upon the subject was continued, tests being run upon vertical lines, horizontal lines, lines of varying pitch and lines of varying size. The program of the work as presented in this report includes the following:

1. Critical velocity of vertical pipes.
2. Critical velocity of horizontal pipes.
3. Critical velocity of pipes, all degrees of pitch.
4. Effect of length upon critical velocity both vertical and horizontal lines.
5. Effect of size of pipe upon critical velocity, both vertical and horizontal lines.
6. Effect of type of entrance upon critical velocity.

It was first deemed advisable to determine if possible by eyesight, the phenomena and effects taking place in a tube in which steam was traveling in one direction and condensate in the opposite, and to note if any relation existed between the visually observed results and those brought out by the pressure velocity curves. Accordingly a thorough series of tests was run using glass tubes of various sizes and forms.

The equipment used in all the experiments of this report was substantially the same as described in the previous report. Briefly, steam generated in a house heat-

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ing boiler was sent through a 6 in. header, through the pipe being tested and into a radiator, whose sole function was to condense all steam delivered to it. The condensation returned by gravity through the test pipe, through a water seal and into a closed bucket where it was weighed at regular intervals. By a special arrangement the pressure in the header was kept constant for each test. The

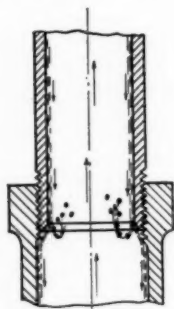


FIG. 1. FIRST POINT OF INTERFERENCE
AT PIPE ENTRANCE

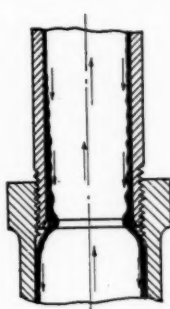


FIG. 2. SECOND POINT OF INTERFERENCE
AT PIPE ENTRANCE

FIG. 1. FIRST POINT OF INTERFERENCE AT PIPE ENTRANCE

FIG. 2. SECOND POINT OF INTERFERENCE AT PIPE ENTRANCE

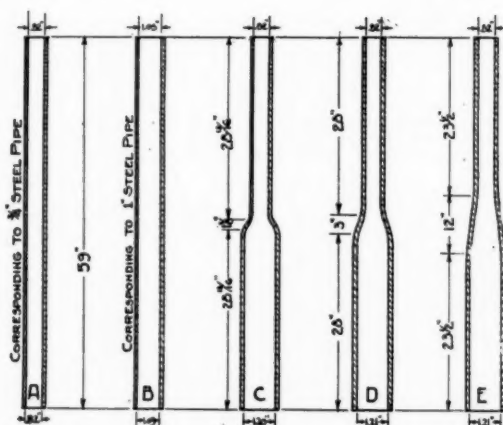


FIG. 3. GLASS TUBES USED TO DETERMINE EFFECT OF
ENTRANCE

final results were plotted using header pressures as abscissae and either condensation in pounds per hour or velocity of steam in feet per second as ordinates.

The range of header pressures used in our experiments varied from 0 to approximately 4 in. of water column. However, and it is important that this point be brought out, the results of all these tests are applicable to any pressures used in low

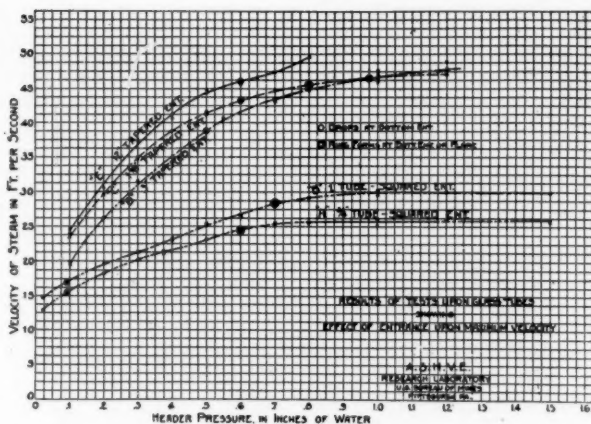


FIG. 4. RESULTS OF TESTS UPON GLASS TUBES WITH VARIOUS ENTRANCES

pressure heating. Flow of steam in a given pipe is dependent upon the pressure drop between the two ends of the pipe. This pressure drop is very small, probably never exceeding 3 to 4 in. of water (2 oz.) under any conditions, and is generally smaller. When a system is being operated at 5 lb. boiler pressure, the pressure existing in the radiators is probably very close to 5 lb. In other words, the pressure differences existing in the risers in our tests, were probably the same as those that exist in the usual heating system.

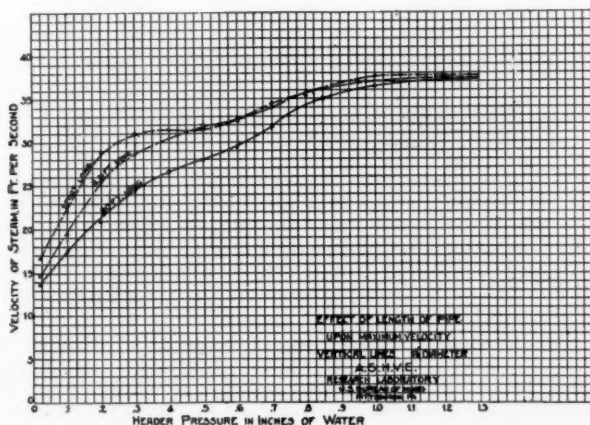


FIG. 5. EFFECT OF LENGTH OF PIPE UPON MAXIMUM VELOCITY

Flow of Steam in Vertical Glass Tubes

The results of the tests upon the vertical glass tubes brought out some interesting features and some points that are fundamental. In order that there shall be a clear understanding of the effects taking place in a tube, when steam is sent up, these results are discussed in detail.

From a visual viewpoint the phenomenon taking place in the tube is as follows: At very low header pressures, the condensate returns down the sides of the tube in smooth and even streams. The steam flows up the center of the tube. There is no noticeable interference between the downcoming condensate and upgoing

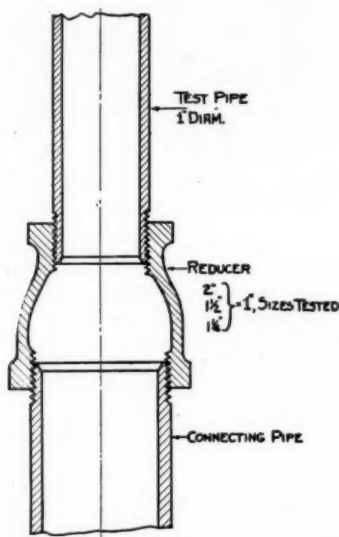


FIG. 6. ARRANGEMENT OF CONNECTIONS FOR TESTS ON EFFECT OF REDUCERS

steam. As the pressure is increased slowly there is an increase in the velocity and amount of returning condensate. Soon a header pressure is reached at which the velocity of the steam is sufficiently high so that the drops of condensate falling off the entrance of the tube are caught by the steam and shot back into the tube. This is the first point of noticeable interference. All along the sides of the tube, however, condensate is still coming down smoothly.

As the header pressure is further increased the drops shot back by the steam into the tube at the entrance increase in number and in intensity, shooting higher and higher up the tube. Gradually there is a change or blending from individual drops to whole masses of water in the form of a ring extending around the sides of the tube, at the entrance. There is also an increase in intensity of flow of the condensate along the sides of the tube; the flow being in the form of distinct waves.

A pressure is reached at which the drops have all disappeared and there is practically a solid ring of water in considerable agitation moving up and down at the entrance. This appears to be the second significant point.

As the pressure is still further increased the ring of water is agitated through a greater length of the tube, the downcoming condensate becoming more and more violent and appearing in greater rings or waves of water all around the tube. If the pressure is made high enough the rings or waves of water are carried to the top of the tube. A point is then reached at which all the condensate is suddenly shot up the tube and back into the radiator. The tube is now clear, that is, there is

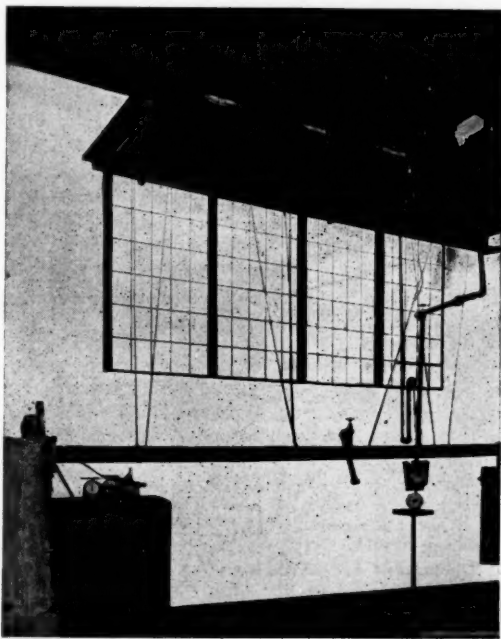


FIG. 7. APPARATUS USED IN CRITICAL VELOCITY TESTS

no condensate coming down. This apparently is the third significant point. If we should continue to run at this pressure without further increase or decrease the steam would continue to flow up the tube with no returning condensate for an extended period. But suddenly for a very short period the condensate would return at full pipe capacity.

The phenomena as presented above apply especially to the straight tubes, that is the tubes with the straight square or reamed entrance. There may be certain slight variations if the type of entrance is somewhat changed, but in general the phenomena will remain the same.

The one outstanding feature in the visual tests as thus far presented is the fact

that all points of significance seem to occur around the entrance of the tube. This suggested the advisability of running some tests on glass tubes, in which the type of entrance was varied. Fig. 3 shows the group of tubes tested. It can be seen that the essential idea was to create a gradually tapered entrance to the tube, so that the condensate traveling down the tube would have no abrupt or sharp change in direction at which to meet the upgoing steam.

Let us now attempt to tie up the results of the experiments upon these tubes as shown by the pressure velocity curve, with the visual observations. The results of tests on the glass tubes are shown in Fig. 4. The following points seem to be of significance.

1. The curve (velocity against header pressure) rises rapidly until it reaches the point where drops of condensate begin to shoot back into the tube. At this point there is a marked flattening out of the curve, that is the velocity still continues to increase but

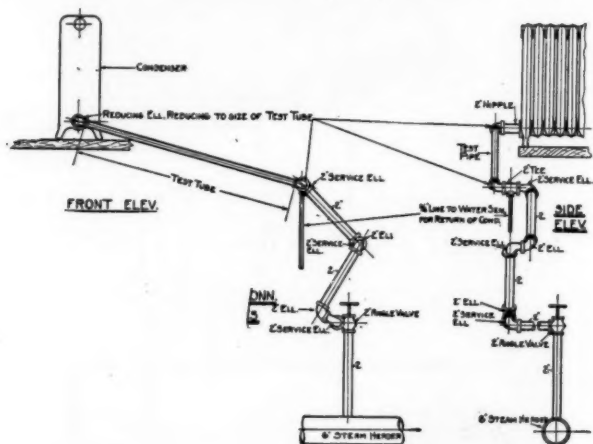


FIG. 8. DIAGRAM OF APPARATUS FOR—TESTS ON HORIZONTAL LINES

at a considerably decreased rate. The point at which drops start shooting back into the tube is designated on the curves.

Flow of steam between two points of a system is due to the pressure difference between these two points, let us say, pressure difference between the entrance to the riser and the entrance to the radiator. As the header pressure is increased the pressure drop is increased and thus the flow of steam is increased. However, when drops of condensate begin to show around the entrance, part of the increased pressure is required to overcome the increased friction caused by the greater number of drops. There is therefore a flattening of the curve as the pressure increases.

2. As the point of the curve is reached where the ring of water shows around the entrance, it is noticed that the curve has become practically horizontal, that is, further increase in header pressure causes no change in the velocity. Note the distinction in effect between the drops of water at the entrance and the ring of water. The drops of water merely cause a slight flattening of the curve, the ring of water flattens the curve entirely.

Again from the point of pressure drop, it would seem that as the pressure at which the ring appears is reached, any increase in header pressure and thus in pressure at the entrance of the tube is required to overcome the increase in resistance caused by the ring

of water, by the decrease in effective area of the pipe, and by the increase in violence of the waves being carried up the tube. Thus the flow of steam remains constant.

3. As the header pressure continues to increase, a point is reached where it is sufficiently great to carry all the condensate back into the radiator. This is the point at which intermittent flow starts, and is affected by the length of pipe. The longer the pipes the greater the pressure required to carry all condensation back to the radiator and the longer is the flat portion of the curve. The length of pipe has no other effect upon the curve.

4. In the report previously published, stress was laid upon the first point where interference begins in the tube. The fact that this was a point of significance, together with the fact that it appeared very close to the point where the critical velocity was expected made it seem as if this was the critical point. However, further work upon this phase showed that, while interference commenced at this point, smooth flow continued for some distance beyond. In other words the curve continued to ascend, even though interference took place until the ring of water appeared in the tube.

These new developments seemed to indicate that it was rather the flat portion of the curve, or the maximum velocity that was the critical point looked for. When a study was made of the flow of steam in horizontal lines, the hump or wave portion disappeared.

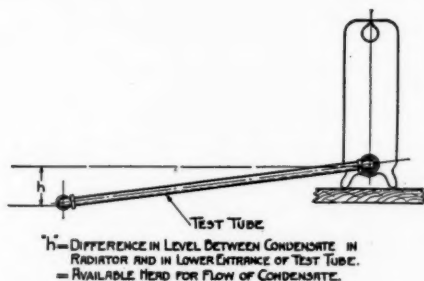


FIG. 9. AVAILABLE HEAD FOR FLOW OF CONDENSATE

In this case, there was no question but that the maximum velocity was also the critical velocity. As the angle of pitch was increased, from a horizontal to a vertical position, there appeared at no time a definite dividing line, between the two distinct phases, critical and maximum velocity.

It was finally decided that until sufficient data had been gathered to analyze definitely and separate each phase and until the effect of each phase had been determined in practice, caution should be used in declaring either to be the critical velocity. The point at which the curve flattens out entirely is really the maximum possible velocity or in other words the maximum velocity which will permit a smooth flow of both the steam and returning condensate. Beyond this point, intermittent flow commences. Just how close to this point it is possible to work in practice can best be determined by some tests on an actual installation, under actual operating conditions.

In the set ups as used in the laboratory no audible surging or water hammer was ever noticed. If the point on the curves at which these noises first appear are once determined by tests on an actual installation it will be very easy to determine from the curves just what is the allowable velocity for all lines. In this report, wherever the flat portion of the curve is dealt with, it will be termed "the maximum velocity."

5. The increase in maximum velocity that can be attained by simply changing the entrance to the tube, so that the returning condensate will interfere as little as possible with the upgoing steam, is very marked. Thus for the straight $\frac{3}{4}$ in. tube, with a squared entrance, the maximum velocity is approximately 24 to 26 ft. per second. The maximum velocity for the $\frac{3}{4}$ in. tube with the 12 in. flare is somewhere between 46 to 50 ft. Here is a 100 per cent increase in the maximum possible flow in the same pipe by merely a change of entrance.

6. A slightly different condition is noted for the glass tube with the 12 in. flare. The curve continues to ascend very rapidly until a header pressure of 0.8 in. is reached when there is a sudden and very large decrease in velocity. At this point intermittent flow is reached. Furthermore, slightly below this point, drops appear in the lower or connecting tube and at no time is there any interference noticeable in the small or test tube. It may be noted from the curves that a great increase in velocity results from flaring the entrance of the pipe.

Flow of Steam in Vertical Steel Tubes

To determine whether the results, as obtained for the vertical glass tubes, will also apply to steel pipes, two series of tests were run: 1. To determine the effect of length upon the maximum velocity and 2. to note if it is possible in practice to increase the maximum velocity for a pipe by flaring the entrance.

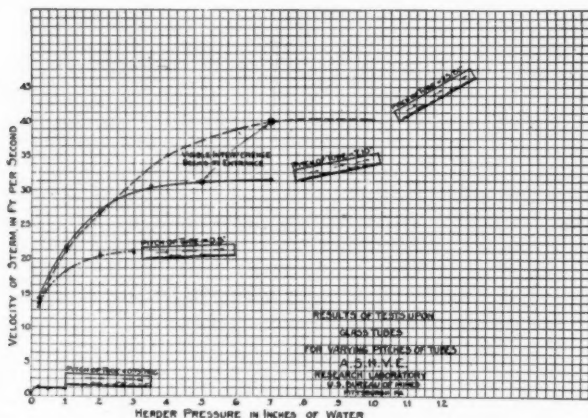


FIG. 10. RESULTS OF TESTS UPON GLASS TUBES—FOR VARYING PITCHES OF TUBES

1. *Effect of Length upon the Maximum Velocity.* Three tubes all of the same internal diameter but of different lengths were tested. The results of these tests are shown in Fig. 5. The velocity of the steam is plotted against the header pressure. The essential feature of these tests is, that the curves all finally arrive at the same maximum velocity. The paths of the curves vary, as might be expected, since the pressure drop for the same header pressure varies with the length of pipe. However, as the header pressure is increased, all three curves tend to draw together until at a pressure of 1.2 in. they are all practically the same. In like manner it was proved several times during the investigation that the maximum velocity is in no way affected by the length of the pipe.

2. Having demonstrated with the glass tubes that the maximum velocity is purely an entrance effect, and that it is possible to change it by changing the size and shape of entrance, an attempt was made to increase the capacity of steel pipes in the same way. Tests were run upon a 1 in. steel riser whose steam entrance was connected to a larger size pipe by means of a reducer. The purpose was to obtain a gradual and rather flaring entrance to the pipe. Fig. 6 shows the typical connections used. It was not possible to increase the maximum velocity beyond that of the straight reamed pipe, the chief reason being, that with the type of fittings found on the market and used in our experiments, it was impossible to do away with the abruptness of the entrance. If a fitting

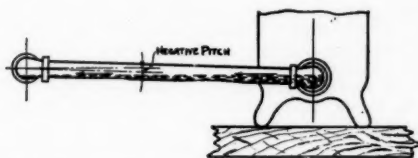


FIG. 11. PILING OF WATER IN TEST TUBE FOR NEGATIVE PITCHES

was used giving a gradually tapering entrance, as used in the glass experiments, the maximum velocity could probably be increased.

Flow of Steam in Horizontal Glass Tubes

Horizontal lines as used in this paper includes not only pipes in an absolutely horizontal position, but also those that are pitched at various angles with the horizontal. In this report the angle of pitch of the pipe is given either in degrees or in inches per 10 ft.

Tests were made on a glass tube in a horizontal position and at several different pitches. Fig. 7 shows a photograph and Fig. 8 a diagram of the typical arrangement of the piping used in all horizontal work. By installing a swing joint in the connections running to the test tubes, it was possible to obtain a large variation in the angles of pitch of the test tubes without disconnecting the system.

In observing the visual phenomena occurring when steam was sent through a horizontal tube three distinct phases manifested themselves. *First*, when the pitch of the pipe was less than about 2 deg., *Second*, from 2 to 90 deg., and *Third*, negative pitches.

1. *Positive Pitches under 2 Deg.* At very low pressures (0.02 in. etc.) condensate travels on the bottom of the tube in a smooth stream. As the pressure is increased (0.08 in. to 0.10 in.) regular waves appear on top of the condensate, which travel toward

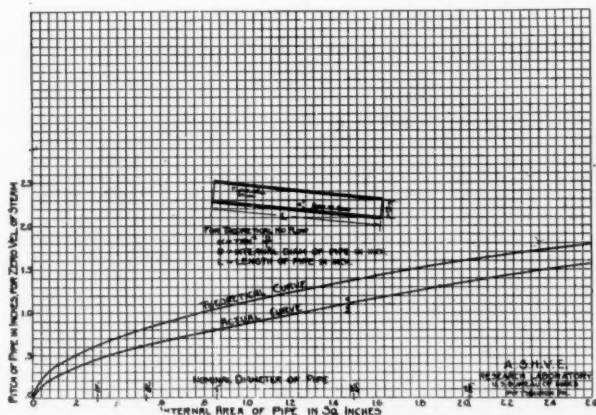


FIG. 12. PITCH OF PIPE AT ZERO VELOCITY OF STEAM

the radiator. On close observation it is found that particles of dirt at the bottom of the tube travel away from the radiator. Particles of dirt at the top of condensate stream travel toward radiator. However, the condensate coming down through the seal is regular and steady. The waves are slightly more pronounced at the tube entrance than at the radiator end of the pipe. As the pressure increases still further, the maximum intensity of the waves shift from the tube entrance to the radiator entrance. At this point the velocity curve becomes practically flat.

This flow differed from that in the vertical tubes, first in that at no time was there any interference at the tube entrance, and second, in that no point of intermittent flow was reached. If the pressure was raised sufficiently, the condensate was all shot back into the radiator. If sufficient time was allowed for the condensate in the radiator to build up a head to overcome the increased head of the steam pressure, the condensate began to flow again in a continuous and uniform stream.

The flow of the condensate from the radiator is due to the difference in the levels of the two ends of the pipe. When this head h is very small, as for very small pitches, the same velocity of steam produces greater waves at the surface of the returning condensate.

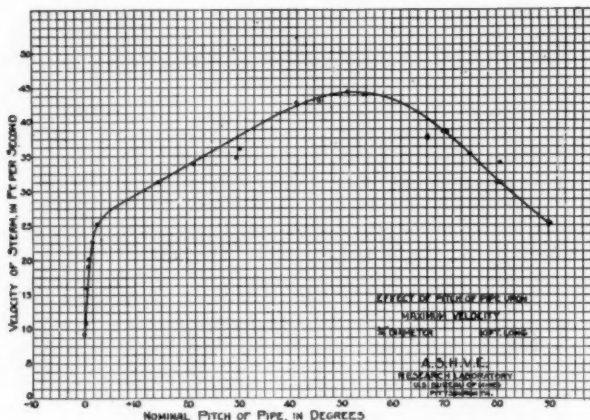


FIG. 13. EFFECT OF PITCH OF PIPE UPON MAXIMUM VELOCITY— $3/4$ " DIAM.

The friction between the steam and returning condensate is then much greater. The effective area of the pipe for the flow of steam is reduced by these waves thus reducing the maximum flow obtainable.

2. *Pitch Varying from 2 to 90 Deg.* When the pitch of the pipe is increased beyond 2 deg. there is a slight change in the observed effect. As the pitch is increased the velocity of the condensate is greatly increased and for the same header pressures, the flow is smooth and steady and there are no waves formed.

Further, since the velocity of the condensate is greater, the actual thickness of the film of water at the bottom of the tube is less. This together with the fact that no waves are formed means that there is a greater effective area for the flow of steam. Consequently the velocity of the steam is increased. For example at 0.10 in. header pressure, Fig. 10, the velocity in the tube is 17.3 ft. per second for a pitch of 0.50 deg. and 21.0 ft. per second for a pitch of 7.10 deg., an increase of over 20 per cent.

It should be stated that unless specifically mentioned the velocities given in this paper were determined by dividing the total volume of steam passing through the pipe by the actual internal area of the pipe. Therefore the values given are always too low since water is at all times present in the pipe and reduces somewhat the area available for steam flow. When the water returns in a smooth stream the reduction in area is small and therefore the difference between the actual and calculated values is small.

The greater the velocity of the returned condensate the less the reduction in area and therefore the smaller the difference between the actual and given velocity. When drops of water are thrown up into the tube, or when waves or any other disturbance of the condensate takes place the free or effective area of the pipe is reduced to a greater extent and in such cases the velocity given may be much lower than the actual velocity of the steam at the point of such disturbances.

As the pitch of the pipe is increased a point is reached at which there is evidence of a slight interference at the tube entrance, a shooting back of an occasional drop of water in the tube, and the formation of waves for a short distance. It is significant that at this point the curve becomes practically horizontal. It may be safe to assume here, that the entrance of the tube is now affecting the results. Furthermore, as will be shown later for steel tubes, if the length of the pipe is increased the results will not change measurably. The two main factors affecting the steam velocity in pipes of this pitch are the angle of pitch available for the flow of condensate and entrance interference.

From 2 deg. to approximately 40 deg. the effect of both factors is in evidence. The effect of the entrance is to retard the maximum capacity and the effect of the pitch is to increase it. The net result is an increase. Thus for the one inch glass tube, Fig. 10,

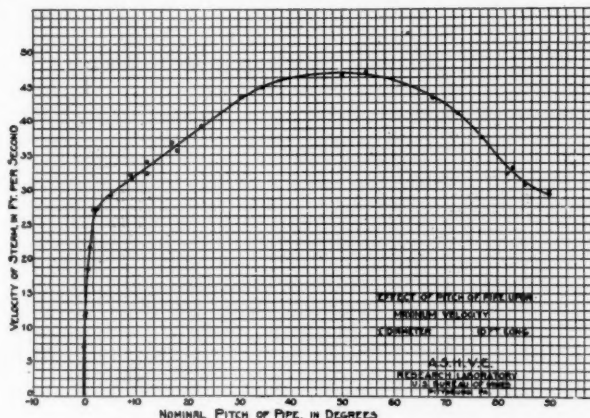


FIG. 14. EFFECT OF PITCH OF PIPE UPON MAXIMUM VELOCITY—1" DIAM.

the maximum velocity at a pitch of 7.1 deg. is approximately 32 ft. per second. At a pitch of 25.8 deg., the maximum velocity has increased to 40.1 ft. per second.

From 40 to 55 deg. the effect of pitch is increasing and the effect of entrance is increasing also, the net result being a flattening of the maximum velocity curve. As the pitch passes beyond 55 to 60 deg. entrance effects are increasing rapidly, while the effect due to the pitch is becoming practically constant. As a consequence the curve is dropping rapidly. Thus, as the pitch passes around the quadrant from 0 to 90 deg. there is a complete change of factors affecting the velocity, from the length of pipe, through the angle of pitch and finally to a purely entrance factor.

3. *Negative Pitch.* When the pitch of the tube is negative we find first that before any flow takes place, water must first be built up in the lower end of the pipe to overcome this negative pitch. The flow of condensate will then be due not to the difference in tube levels, but merely to the difference in water level at the radiator and steam entrance ends of the tube. This difference in levels is very small. Furthermore due to the water piling up at the radiator entrance, the effective area for the flow of steam is greatly reduced.

However up to a certain negative pitch there will be a flow of condensate, decreasing as the negative pitch is increased. This pitch is reached theoretically when the tangent of the angle of pitch (α) is equal to $\frac{\text{Internal Diameter of Pipe}}{\text{Length of Pipe}}$. Practically due to

experimental difficulties, it is impossible to approach this pitch very closely. Fig. 12 will give an idea as to how closely the theoretical pitch is approached.

The pitch of the pipe in inches is plotted against the area and diameter of the pipe, for both the theoretical and experimental points. Thus for the 1 in. pipe the theoretical negative pitch for no flow is the internal diameter of the pipe, which is 1.046 in. The actual negative pitch as found experimentally for the one inch pipe is 0.80 in. approximately.

Flow of Steam in Horizontal Steel Tubes

At the conclusion of the tests upon the horizontal glass tubes, a series of tests was run upon steel tubes of various sizes and lengths, and with various angles of pitch, to determine if possible all factors entering into the flow of steam in inclined pipes.

1. *Effect of Pitch upon the Maximum Velocity.* The results of all the tests are plotted in Figs. 13, 14, and 15. In each case the velocity in feet per second is plotted against

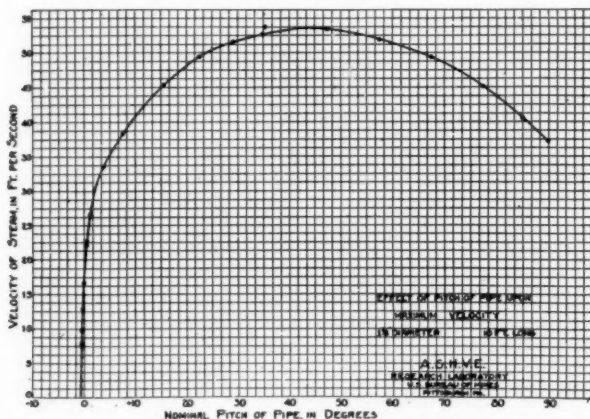


FIG. 15. EFFECT OF PITCH OF PIPE UPON MAXIMUM VELOCITY—1 1/4" DIAM.

the nominal pitch of the pipe in degrees from the horizontal. These curves indicate:

- For small pitches from 0 to 2 deg. the maximum velocity increases rapidly with the pitch. During this period, the maximum velocity varies considerably with the length of pipe. This point will be further discussed later.
- As the pitch increased beyond 2 deg. the maximum velocity continues to increase, almost directly proportional to the angle of pitch until the maximum is reached at an angle of 35 to 50 deg., depending on the size of pipe.

Beyond a pitch of 2 deg. the maximum velocity is practically independent of the length of pipe.

- From 50 to 90 deg., the effect of the entrance of the pipe continues to increase and the maximum velocity decreases, until at a pitch of 90 deg. or a vertical pipe, the entrance to the pipe is the main factor affecting the maximum velocity.

- As commonly expressed the nominal pitch is the angle formed by a horizontal line and a line parallel to the center line of the pipe. The absolute pitch however, is greater than the nominal pitch by an amount which approaches the internal diameter of the pipe as the angle of pitch is decreased.

If ϕ is the nominal pitch of the pipe then

$$\alpha = \sin^{-1} \left(\frac{h}{L} \right)$$

where L = length of pipe in inches.

h = pitch of pipe in inches.

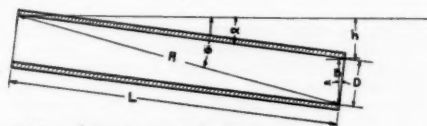
The absolute pitch is

$$\begin{aligned}\phi &= \sin^{-1} \left(\frac{h + B}{A} \right) \\ &= \sin^{-1} \left(\frac{h + D \cos \alpha}{\sqrt{L^2 + D^2}} \right)\end{aligned}$$

For example if a 1 in. pipe 10 ft. long is given a nominal pitch of 2 in. then $\sin \alpha = \frac{2}{120} = 0.01667$ and, the nominal angle of pitch = 0.95 deg.

The absolute pitch, $\phi = \sin^{-1} \left(\frac{2 + 1.04 \cos (0.95 \text{ deg.})}{\sqrt{120^2 + 1.04^2}} \right) = 1.45 \text{ deg.}$

As might be expected from the above, when a pipe is tested in a horizontal position or at a nominal pitch of 0 deg., the velocity of flow is not zero, but is considerably greater, depending upon the length of the pipe and diameter of opening. This is shown graphically by Curve B on Fig. 19.



α = NOMINAL PITCH OF PIPE IN DEGREES

$$= \sin^{-1} \frac{h}{L}$$

h = PITCH OF PIPE IN INCHES

L = LENGTH OF PIPE IN INCHES

ϕ = ABSOLUTE PITCH OF PIPE IN DEGREES

$$= \sin^{-1} \frac{h + B}{A}$$

BUT $D = \frac{D \cos \alpha}{\cos \alpha}$

AND $A = \sqrt{L^2 + D^2}$

$$\text{THEN } \phi = \sin^{-1} \frac{h + D \cos \alpha}{\sqrt{L^2 + D^2}}$$

D = INTERNAL DIAMETER OF PIPE IN INCHES

FIG. 16. VARIATION BETWEEN NOMINAL AND ABSOLUTE PITCH

e. The velocity of flow becomes zero, when the nominal pitch as expressed in inches for the actual length of pipe becomes a negative value somewhat smaller than the internal diameter of the pipe, or in other words when the absolute pitch is nearly zero. This was brought out under the subject of horizontal glass tubes.

2. *Effect of Length of Pipe upon Maximum Velocity. Horizontal Lines.* For very small pitches of pipe, up to about 2 deg. the maximum velocity is greatly affected by the length of the pipe. A series of tests upon four different lengths of $1\frac{1}{4}$ in. pipe shows this clearly. The results are shown in Figs. 17 and 18. For example, from Fig. 17 the velocity for the 10 ft. pipe was found to be approximately 16 ft. per second at zero pitch, the velocity for the one foot of pipe (the other extreme tested) was found to be 25.5 ft. or a gain of close to 60 per cent. As the pitch is increased positively, all curves tend to draw together, until for a pitch of about 2 deg. or 3.5 in. per 10 ft. the results for all lengths are the same. Above this point the length does not measurably affect the maximum velocity.

Fig. 18 will bring out the effect of length a little more clearly. In this figure the velocity is plotted against the length of the pipe and also against the angle of opening which is a function of the length of pipe, for a nominal pitch of zero degrees. The maximum velocity increases very rapidly as the length decreases. Theoretically it will continue to increase indefinitely as the length of pipe decreases. Experimentally, however, if it was possible to test a piece sufficiently short the limiting conditions would approach that of a thin plate orifice, whose opening was the same as the diameter of the pipe tested.

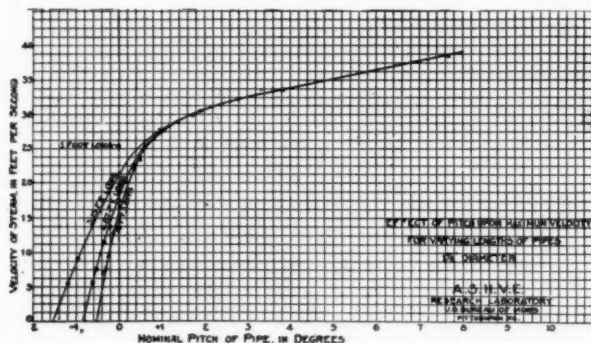


FIG. 17. EFFECT OF PITCH UPON MAXIMUM VELOCITY—VARYING LENGTHS OF PIPES— $1\frac{1}{4}$ " DIAM.

3. *Effect of Size of Pipe upon the Maximum Velocity.* In general the maximum velocity increases as the size of pipe increases for sizes of pipes up to about $1\frac{1}{4}$ in. diameter. Beyond that the maximum velocity is practically independent of pipe size. This relation seems to hold true regardless of the pitch of the pipe. Referring to Fig. 19 the maximum velocity and capacity curves are shown for both horizontal and vertical lines. All curves are plotted against the area of the pipe as abscissae. It is interesting to note that the general form of the curves is the same for both the horizontal and vertical lines. In both cases the velocity increases as the size of pipe increases for small sizes of pipes, and tends to approach a constant value for sizes of pipes above $1\frac{1}{4}$ in. diameter.

Practical Application

It has been the object of this investigation to present a complete story of the flow of steam in pipes. The essential purpose of these experiments is to obtain a thorough understanding and complete knowledge of the factors entering into,

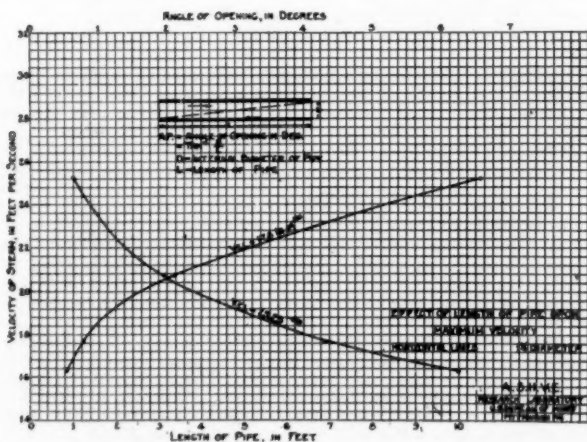


FIG. 18. EFFECT OF LENGTH UPON MAXIMUM VELOCITY— $1\frac{1}{4}$ " DIAM.

TABLE 1. CAPACITIES OF ONE PIPE LINES—VERTICAL AND HORIZONTAL
Allowable Velocity in Feet per Second

Nominal diam. of pipe in in.	Cond. in lb. per hr.	18			22			Pitch of pipe in. per 10 ft.
		B.t.u. loss per hr.	Sq. ft. rad. based on 240 B.t.u.	Pitch of pipe in./10 ft.	B.t.u. loss per hr.	Sq. ft. rad. based on 240 B.t.u.	Pitch of pipe in. per 10 ft.	
$\frac{3}{4}$	8.9	8,640	36.0	1.00	10.9	10,580	44.1	2.6 to 3.0
1	14.5	14,070	58.6	0.75	17.8	17,270	72.0	1.7 to 2.0
$1\frac{1}{4}$	25.2	24,460	101.9	0.50	30.8	29,890	124.5	1.00
$1\frac{1}{2}$	34.2	33,190	138.1	0.50	41.8	40,560	169.0	1.00
2	56.5	54,830	228.4	0.50	69.0	66,960	279.0	1.00

and the effects taking place when steam is sent through a pipe in which there is a counterflow of condensate. It is only by understanding these factors that a proper application of them can be made in practice.

From the point of view of the practical applications of the results it is of interest to learn: *First*, the pitch a pipe must be given to obtain a certain velocity; *Second*, the capacity of pipes of various pitches; *Third*, the best working pitches under average conditions.

TABLE 2. CAPACITIES OF ONE PIPE LINES AT VARIOUS PITCHES
Pitch of Pipe in Inches per 10 Feet

Nominal diam. of pipe in in.	Cond., lb./hr.	$\frac{1}{4}$			$\frac{1}{2}$		
		B.t.u. loss/hr.	Sq. ft. rad. based on 240 B.t.u.	Pitch in./10 ft.	B.t.u. loss/hr.	Sq. ft. rad. based on 240 B.t.u.	Pitch in./10 ft.
$\frac{3}{4}$	6.19	6,010	25.0	7.48	7,260	30.3	
1	11.32	10,990	45.8	13.02	12,630	52.6	
$1\frac{1}{4}$	25.95	25,180	104.9	29.00	28,140	117.2	
$1\frac{1}{2}$	35.25	34,210	142.6	39.30	38,140	159.0	
2	58.40	56,670	236.0	65.20	63,270	263.5	
1							
$\frac{3}{4}$	9.22	8,950	37.3	10.00	9,700	40.4	
1	15.58	15,110	63.0	17.30	16,790	70.0	
$1\frac{1}{4}$	32.93	31,950	133.0	35.75	34,690	144.5	
$1\frac{1}{2}$	44.80	43,470	181.0	48.60	47,160	196.5	
2	74.10	71,910	299.5	80.50	78,120	325.5	
2							
$\frac{3}{4}$	10.50	10,190	42.5	11.40	11,060	46.1	
1	18.60	18,050	75.2	20.55	19,940	83.0	
$1\frac{1}{4}$	38.10	36,970	154.0	40.80	39,500	165.0	
$1\frac{1}{2}$	51.80	50,270	209.3	55.50	53,860	224.0	
2	85.70	83,160	346.5	91.90	89,180	371.5	
3							
$\frac{3}{4}$	11.74	11,390	47.5	12.19	11,830	49.3	
1	21.75	21,110	87.9	22.31	21,650	90.2	
$1\frac{1}{4}$	42.68	41,420	172.6	44.07	42,760	178.2	
$1\frac{1}{2}$	58.10	56,350	248.4	60.00	58,220	242.6	
2	96.80	93,240	388.4	99.21	96,270	401.1	

1. In the report published in March, 1923, a table was given showing the capacities of vertical pipes of various sizes for, first, a velocity of 22 ft. per second, which is really the minimum critical velocity obtained and second, a velocity of 18 ft. per second, which permits a factor of safety of approximately 20 per cent. It is obvious that if either of the tables are to be used for determining the size of vertical lines, then to obtain smooth and even flow horizontal lines must be so designed as to give the same capacity.

Table 1 has been designed with the above in mind. With the exception of the column showing the pitch of the pipe in inches per 10 ft., the table is the same as was given in the last report. The values in the first half are based upon a velocity of 18 ft. per second and those in the second half on a velocity of 22 ft. per second. The table gives the capacity

of the pipe in pounds of steam per hour, the B.t.u. supplied per hour, the square feet of radiation based on 240 B.t.u. per sq. ft., and the necessary pitch to be given a pipe to obtain the desired capacity.

For example it is desired to feed a radiator of 100 sq. ft. of radiation without exceeding a velocity of 18 ft. per second. From our table we find that a $1\frac{1}{4}$ in. pipe will take care of 102 sq. ft. of radiation. If this pipe is to be in a horizontal position such as a runout, the last column of the table shows that the pipe must be pitched 0.50 in. per 10 ft. Pitching the pipe more will be of no value since the velocity in the vertical line will still be 18 ft. Pitching the pipe less will cut down the carrying capacity of that pipe and thus decrease the capacity of the radiator.

2. From Figs. 13, 14 and 15 it may be seen that the capacity of a pipe increases very rapidly for pitches up to about 2 deg. or about 4 in. per 10 ft. The capacity continues to increase beyond that point also, but at a somewhat reduced rate. From the results of the experiments, the limiting pitch of a pipe is approximately 45 deg. If it were possible and practical to install a system using only lines pitched at 45 deg., the capacity of the system would be greatly increased. However, in an actual installation this is not practical, and a pitch of 4 or 5 in. per 10 ft. is probably the maximum allowable.

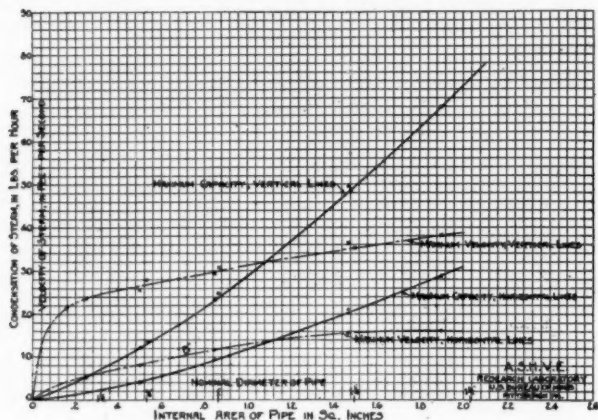


FIG. 19. MAXIMUM CAPACITY AND VELOCITY OF PIPES—FOR VARIOUS SIZES

Table 2 has been designed to show the increase in capacity of a pipe as the pitch is increased. The condensation in pounds per hour, the B.t.u. supplied per hour and the square feet of radiation are all given for pipes of various sizes and pitches. With the exception of the 2 in. pipe these figures are based upon the results given in this report. The figures for the 2 in. pipe are based upon the deduction brought out in this report that above $1\frac{1}{4}$ in. diameter the velocity in a pipe is independent of the size of the pipe.

For example it is desired to supply steam to 175 square feet of radiation. This may be accomplished by using a large size pipe with a small pitch or a smaller size pipe with a greater pitch. In industrial and other buildings where head room is available, and there is no other objection to using large pitches, the size of pipe can be materially reduced. Thus for the case assumed, it may be found from Table 2 that $1\frac{1}{4}$ in. pipe with 1 in. pitch per 10 ft. will supply 181 square feet of radiation. If conditions are such that the pitch may be greatly increased, we find that $1\frac{1}{4}$ in. pipe with a pitch of 5.00 in. per 10 ft will feed 178 sq. ft. of radiation. Increasing the pitch reduces the size of pipe required.

3. Table 1 indicates that the pitch required for a certain velocity is greater for small sizes of pipe. It may be desired to know what general pitch is applicable to average conditions. A pitch of $1\frac{1}{4}$ in. per 10 ft. will give a velocity ranging from 20 ft. per second for $\frac{3}{4}$ in. size to 26 ft. per second for 2 in. size. In average conditions this pitch would be about the most commonly used. The portion of Table 1 figured on a velocity

of 22 ft. per second would apply very closely to this condition. The lower limit of pitch advisable is about as indicated in the first half of the table. This gives a variation in velocity from 18 to 24 ft. per second.

4. Tables 1 and 2 are based upon 10 ft. lengths of pipe. For vertical lines the velocity is independent of the length of pipe. For horizontal lines of very small pitch the velocity decreases as the length increases. Thus the velocity at 0 deg. pitch for $1\frac{1}{4}$ in. pipe 10 ft. long is 16.3 ft. per second. If the pipe is 50 ft. long, the velocity is 12.5 ft. per second, a decrease of about 25 per cent. It is therefore important that with lengths of pipe greater than 10 ft., allowance shall be made for the decrease in the velocity.

The average decrease in velocity for lengths between 10 and 50 ft. is about 6 per cent per 10 ft.

DISCUSSION

E. S. HALLETT: I hope it will be possible for the Research Laboratory to conduct the experiments along somewhat different lines. Taking into consideration the sizing of horizontal pipes, where the condensation which has to be taken care of results from the supply pipe itself and not from the radiator. We want an answer to this practical question; In a two-pipe system, what is the proper size of connection from the riser to the radiator when the radiator is 10, 15 or 20 ft. away from the riser?

H. M. HART: I would like to ask if the pipes were reamed in these tables. I assume that they were but I want to make it a matter of record.

T. M. DUGAN: I would like to ask if it wouldn't be advantageous to use recess fitting to overcome the phenomena taking place at the end of the pipe. For it seems from the data presented by the author of the paper that reaming of pipe thus reducing the friction surface is a very important factor.

Hence I believe that the use of recess fittings particularly at the bottom of risers would eliminate this feature entirely at that point.

The many apparent advantages thus derived would readily compensate for the difference in the price of the fittings.

R. G. TAGGART: I might say, in connection with recessed fittings, that we had a job where we tried to use recessed fittings for hot-water heating and the greatest pressure that we could get any manufacturer to guarantee was practically a negligible pressure. I believe the suggestion of increasing the horizontal connection at the lower end of the steam riser, starting at the foot of the riser, has many advantages.

Another factor is this question of burr in a pipe, which reduces the effective size of the pipe. I have had considerable correspondence with an old friend of mine who does steam fitting in the Hague (and I imagine in some other parts of Europe). They have no trouble with burrs. The workmen are trained to remove them and the size of pipes which they use is astonishing as compared with our practice. But when we attempt to eliminate burrs from the pipes, we generally find that the cost of being sure that all of our burrs are removed is more than the cost of using a larger pipe.

Another point is this—is the critical velocity always that thing that determines the size of a steam riser? In my opinion it is not. I, at one time, was connected with a concern that made some considerable experiments on velocities in pipes with glass pipes. We found that when you came to the larger sizes of pipes we could afford to have a certain amount of water held up in the vertical steam riser in order

that the water would agglomerate into large drops, so that the velocities that might be used in 3 or 4 in. pipes were very much greater than the velocities that were practical in 1½ in. pipes. At that time I went through a number of office buildings, checked up the radiation and found that apparently it was this agglomeration into the larger size drops in the three and four inch pipes which did not result in noise but allowed a great deal higher velocity in vertical steam risers than you would expect from these tables. I hope that in extending this work, experiments will be made on some of the larger size pipes.

T. H. IRELAND: On the subject of reaming pipes, I have seen many installations where they have reamed the pipe but the steam fitter has used very heavy dope on the female thread. By female thread I mean the thread within the fitting, thus permitting a bigger obstruction than the burr on the pipe. You will find that one of the greatest causes of repairing a heating system is the dope after it has become hard and solid, going down and accumulating between the disc and set of the valves.

J. A. DONNELLY: I want to speak of the table. If you take the amounts of radiation that are on these sizes of pipe and say those are safe for starting loads, I think you go a little beyond what is safe from our present practice, in fact, I think this table should be used with a great deal of discretion. I doubt whether very many members of the Society will put this table into use. I would, with a great deal of caution, because I know under some conditions trouble has ensued with amounts of radiation somewhat less than those shown. It might be due to slight priming of the boiler, due to dirty water or defective boiler design, or due to too limited capacity of circulation. In other words, there is a critical capacity of circulation within the boiler itself. It might be due to many other causes which could only be obtained perhaps in field work.

It has always been my idea that the safest thing is to adopt research where it agrees with our previous experience. When it changes our previous experience it will change it but very slowly, because men are loathe to increase capacities and to deviate from what they have done in the past, and rightly so.

F. C. HOUGHTEN: All data given in these tables are for reamed pipe. A question in regard to recessed fittings was asked. I am not familiar with these fittings, but I will say that if it is desirable you can increase the capacity of a riser by gradually decreasing its entrance. However, there is some doubt as to whether that is economically practical. As shown in the curves, tapering at the lower end of a pipe to a larger size, may double its capacity.

H. M. HART: In this question of reduction in the riser slightly above the elbow, page 154, it gives dimensions of about 23 to 28 in. from the base to that point where the reduction is made. I would like to ask if that was tried out at different distances from the top of the elbow. Another point is that in single pipe, up-feed steam heating, I think it is general practice to make the horizontal branch to the riser one size larger than the riser, which calls for a reduced elbow. I don't believe that a straight elbow and a reducer such as is illustrated here would increase the expense to a very large extent.

F. C. HOUGHTEN: The taperings in the pipe were all at the same height. However, that point represents the bottom of the pipe. The large section of the pipe is so large that the length of it does not enter in. Tapering the entrance, will give an increased capacity, as Mr. Hart mentioned. However, I do not believe a reducing couple with a reamed pipe, made as the couplings are now, will answer that purpose. As brought out in this paper, that was tried but without success. I believe, however, that if an iron fitting was turned down so you had a slow taper, it would work. The tapered pipes shown in the paper are all made of glass.

SIMULTANEOUS FLOW OF WATER AND AIR IN PIPES

By L. S. O'BANNON,¹ LEXINGTON, KY.

MEMBER

THE condition of flow in the dry return of a vapor heating system is that of simultaneous flow of water and air. The water flows in the bottom of the pipe, down grade, due, primarily, to the pull of gravity. The air flows in the top of the pipe and may accelerate or impede the flow of water.

It has been generally recognized that there exists a certain critical relation between the factors affecting the continuity of flow in the pipe—a critical condition, below which the flow of water and air remains steady, and beyond which the flow is intermittent, impulsive, turbulent, or at any rate, not conducive to the noiseless and efficient operation of a steam heating system.

The object of the experiments described in this paper has been to find the capacity of pipes for carrying water and air simultaneously, within the conditions of steady flow. So far, the capacity of a 1 in. pipe, having a slope of 1 in. in 10 ft., has been determined for both parallel and counter flow.

The apparatus used in these experiments Fig. 1 shows the arrangement for testing. A 1 in. pipe, 20 ft. long, with one end 2 in. higher than the other, was mounted between two reservoirs. A German silver wire, stretched taut, parallel to, and about 6 in. below the pipe, was used to determine the alignment of the pipe between the reservoirs. Wire hangers, with turnbuckles, supporting the pipe were adjusted so that at no place in the run of the pipe did the distance from the wire vary more than about $\frac{1}{60}$ in. The frame holding the inlet reservoir was supported on adjustable screws. A civil engineer's surveying level, set on a tripod about 15 ft. from the apparatus was used for establishing the slope of the pipe.

Water from a supply tank flowed by gravity into the inlet reservoir. The supply tank had a wide overflow near its top for the purpose of maintaining a constant head. The amount of water let into the reservoir was regulated by a valve in the line connecting the bottom of the supply tank and the bottom of the reservoir. With this arrangement it was possible to maintain a constant head in the reservoir, and therefore, to keep a constant flow of water through the pipe at any particular depth. A micrometer depth gage projecting through the top of

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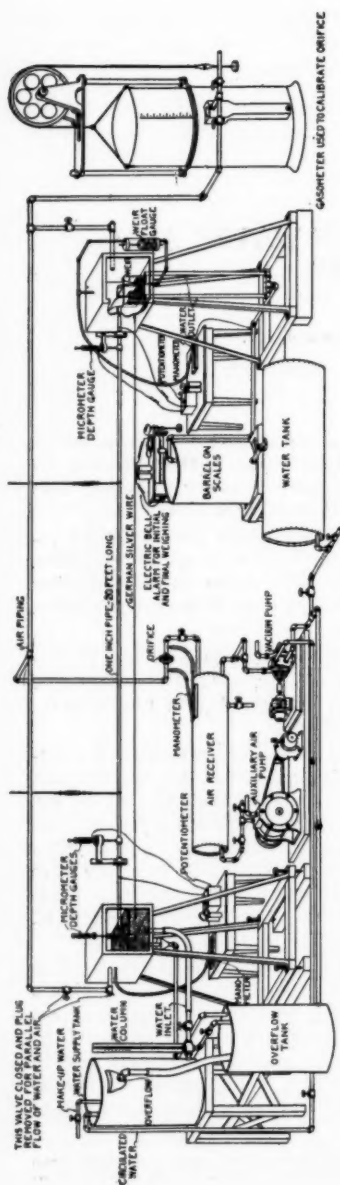
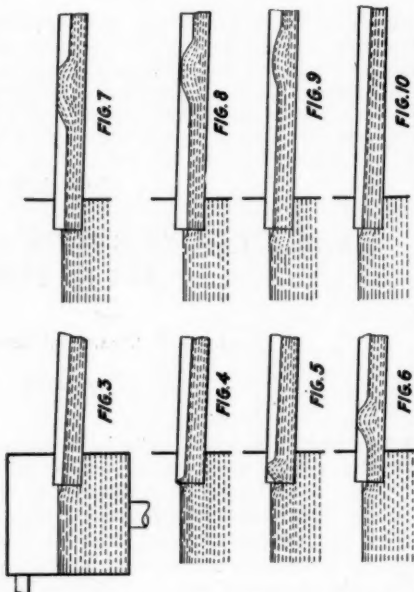


FIG. 1. ARRANGEMENT OF TESTING APPARATUS



FIGS. 3-10. ILLUSTRATE THE DISTURBANCE WHICH TAKES PLACE AT ENTRANCE OF PIPE FOR VARIOUS DEPTHS OF WATER WITHIN

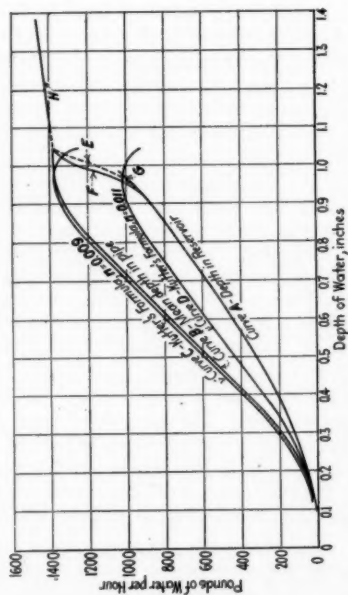


FIG. 2. CURVE SHEET SHOWING RESULTS OF ACTUAL OBSERVED READING

the reservoir was used to obtain the height of the water in the reservoir above the entrance of the pipe.

The water flowed through the pipe into the outlet reservoir through a weir-box and a 30 deg. V-notch weir, and thence into a barrel on scales. The scales were equipped with an electric bell alarm operated by the scale beam. This device was useful for starting and stopping the tests, and for eliminating possible errors in initial and final weighing.

A float gage was attached to the weir-box so that the gage indicated the head of water on the weir and was used in some of the tests for obtaining the rate of flow of the water. In the later experiments, a hydrometric goblet, or orifice bucket, not shown on the drawing, was placed in the position that the barrel and scales now occupy. The hydrometric goblet was found to give more accurate results as it indicated the variation of flow better than the weir float gage, and also, for the reason that different orifices could be used to suit the rate of flow.

The water in the barrel on the scales was emptied into a tank and then returned to the supply tank by means of a motor driven centrifugal pump of the combination water and air type. This pump also returned the water from the overflow tank to the supply tank.

The air system consisted of the combination water and air pump, a receiver, orifice meter, gasometer and the piping connecting these to the reservoirs. The air piping was so arranged that air could be drawn through the test pipe in either direction. In case of parallel flow of water and air the inlet reservoir was open to the atmosphere and the air was exhausted from the outlet reservoir. In case of counter flow the valves were manipulated so as to reverse the flow and the outlet reservoir was opened to the atmosphere. The gasometer was permanently attached to the apparatus and was used for calibrating the orifice meter. It was shut off from the system when tests were being made. An inclined U-tube manometer was used for measuring the differential pressure across the orifice. The orifice meter itself consisted of a thin brass plate containing a $1\frac{1}{2}$ in. round hole, the plate being fastened between flanges connecting 1 and $1\frac{1}{4}$ in. pipes. The pressure drop between the two reservoirs, that is, between the inlet and outlet of the test pipe, was measured by an inclined U-tube manometer.

The depth of the water in the pipe was obtained by means of two micrometer depth gages, one placed one foot from the entrance and the other placed one foot from the outlet. The part of the depth gage projecting into the pipe consisted of a steel needle less than 0.05 in. in diameter. A small hole drilled in the top of the pipe was covered with gasket rubber. The needle was thrust through the rubber thus insulating the needle from the pipe and also preventing the entrance of air. One terminal of a potentiometer, like that used in connection with thermal couples for measuring temperature was grounded to the pipe and the other terminal was connected to the depth gage. The depth gage being insulated from its support, a deflection of the potentiometer occurred only when the needle made contact with the water in the pipe. The method of taking readings was first to raise the needle until contact with the water was broken, and then, to screw down on the depth gage until contact was made as indicated by the movement of the galvanometer pointer. The difference between the observed depth gage reading and the zero reading when the needle touched the bottom of the pipe was considered to be the depth of the water. Each observation may not have been absolutely correct in the sense that it gave the actual mean depth at that cross section of the pipe, because it was impossible to know whether the needle at the instant of the galvanometer de-

flection was making contact with the crest or valley of a wave on the surface of the water. With no air flowing through the pipe the amplitude of a wave would be negligible, but with air flowing the wave motion was exaggerated and the observations might be in error an appreciable amount.

The results and the methods of running the experiments will be described under four headings: 1. The Flow of Water; 2. The Flow of Air; 3. The Parallel Flow of Water and Air; and 4. Counter Flow of Water and Air.

The Flow of Water

Observations were made on the flow of water through the pipe with atmospheric pressure at each end, or, in other words, with no air flowing, for the purpose of determining the rate of flow for different depths in the pipe. Single tests running for a period of about half an hour were made for depths in the pipe ranging from $\frac{1}{10}$ in. to a full pipe. The three depth gages were read every 5 min. The water

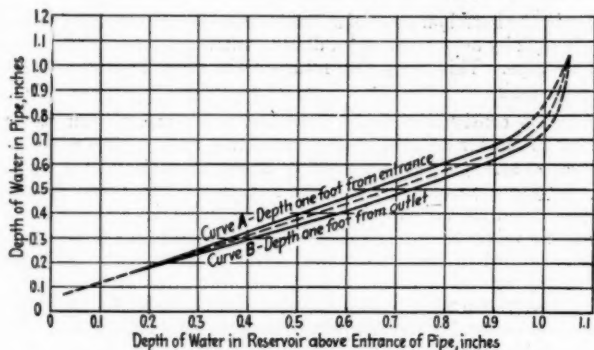


FIG. 11. SHOWING DEPTH OF WATER IN PIPE ONE FOOT FROM INLET AND ONE FOOT FROM OUTLET PLOTTED AGAINST DEPTH OF WATER IN RESERVOIR.

was weighed in the barrel on the scales. The weir gage readings were observed and it was from the data of these tests that a calibration curve for the weir gage was obtained. The most consistent and the most reliable observation made was the depth of the water in the reservoir. This measurement, therefore, has been used as the chief ordinate for illustrating the results.

On the sheet of curves showing pounds of water per hour plotted against depth of water Fig. 2, the curves marked A, F and G were plotted from the results of the actual observed readings of the inlet reservoir depth gage. Curve A, below the intersection of F and G, was very definitely determined. The reason for the divergence of curves F and G may best be explained by reference to Figs. 3-10, which illustrate the disturbance which takes place at the entrance of the pipe for depths of water within about $\frac{2}{10}$ in. of a full pipe. Fig. 3 shows the water flowing steadily from the reservoir into the pipe. The water in the reservoir is higher than the water in the pipe, the difference in head representing that which is required to effect the change of velocity and overcome the resistance at the mouth of the pipe. On account of the velocity of the water changing abruptly from zero to the velocity of the water in the pipe there is considerable disturbance at the entrance. As a

full pipe is approached a wave or ripple may seal the mouth of the pipe as illustrated in Fig. 4. The system then appears to act as a siphon. The wave grows and travels rapidly down the pipe as shown by Fig. 5, Fig. 6, and Fig. 7. The depth of the water ahead of the wave increases and consequently the amount of water discharged from the outlet is increased. The increase in flow causes the head in the reservoir to decrease. This, in turn, will allow water to flow faster from the supply tank into the reservoir. But the pipe is drawing the water from the reservoir at a greater rate than water is being supplied; the water in the pipe behind the traveling wave is lowered and the seal is broken. The depth through the whole system at this moment is slightly less than at the beginning of the disturbance and normal flow then ensues. As the water again approaches a full pipe, the phenomenon may be repeated. There are evidently time and space elements involved. The disturbing wave may last for a fraction of a second or longer and may occur several times in a minute. It may break a few inches from the entrance or it may travel the whole length of the pipe. Naturally, the pipe depth gages with their

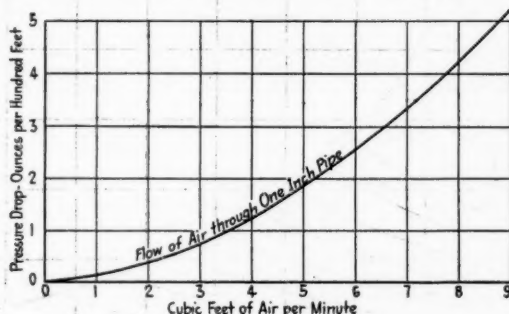


FIG. 12. CURVE SHOWING FLOW OF AIR THROUGH ONE INCH PIPE

electrical connections are of no use when this disturbance begins. The maximum depth measured with some degree of precision with the inlet depth gage was about 0.75 in. and with the outlet depth gage about 0.70 in. The corresponding depth of water in the reservoir was about 0.95 in. above the entrance of the pipe. The inlet depth gage would be the first to become short circuited and then with a slight increase in the reservoir depth the same would happen to the outlet gage.

Referring to Fig. 2 it is evident that the rate of flow of water per hour indicated by curve *F*, corresponding to depths near a full pipe, is greater than what actually should obtain if it were possible to eliminate the disturbance described above. Because, for a fraction of the period of the tests, a part of the pipe was flowing full instead of remaining steady at the apparent depth.

With particular attention to this region of the curve, a few tests were made with conditions maintained as smoothly and regularly as possible. The results are shown by curve *G*. For depths beyond the end of curve *G* the pounds of water per hour checked with the upper curve. It is very probable that curve *G* gives values too low, because in the endeavor to maintain constant conditions and avoid the disturbance at the mouth of the pipe, the investigators would be prone to regulate the supply of water sparingly rather than allow the system to assume its normal bent.

The dotted curve, *E*, was arbitrarily drawn and indicates, by the judgment of the investigators, the true course of the line, if it were possible to avoid the disturbance at the pipe entrance.

Several tests were made with the pipe running full and with the mouth of the pipe fully submerged. The results of these tests are shown by curve *H*. By projecting this line to the ordinate corresponding to a full pipe, 1.05 in., the capacity of the full pipe is determined for the condition of atmospheric pressure at each end and with no additional head beyond that due to gravity and the weight of the water. This rate of flow is about 1390 lb. per hour.

Fig. 11 shows the depth of water in the pipe 1 ft. from the inlet and 1 ft. from the outlet plotted against the depth of the water in the reservoir. The upper dashed portions of the curves are imaginary, since no depth gage readings were obtained

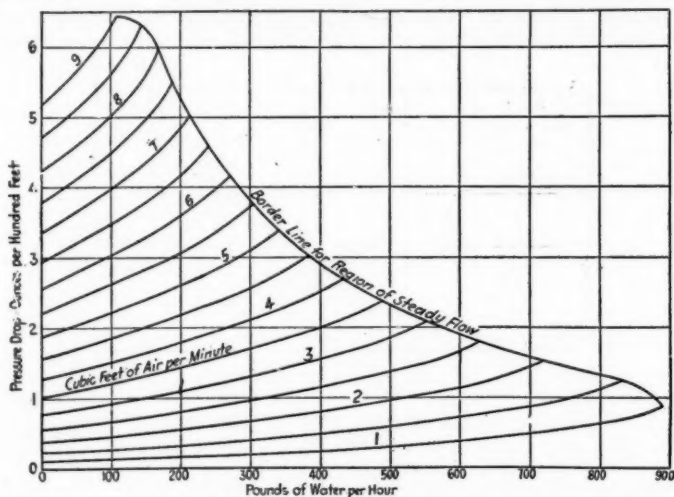


FIG. 13. A CAPACITY CHART FOR SIMULTANEOUS FLOW OF WATER AND AIR IN ONE INCH PIPE—PARALLEL FLOW

higher than indicated by the full lines. The dotted curve is meant to strike an average between the curves *A* and *B*, and gives the mean depth of the water in the pipe. It will be noticed that at the intersection of these curves, the inlet, outlet, and reservoir gages all indicated about the same depth. This means that the acceleration due to gravity was just enough to overcome the frictional resistance of the pipe. Consequently, there is no change in velocity, nor in depth. The reading at this point is about 0.16 in. and the amount of water corresponding to this depth is about 40 lb. per hour.

On curve sheet Fig. 2, the horizontal scale for curve *B* gives the mean depth of the water in the pipe.

While the investigators were preparing for these experiments, Kutter's formula for the flow of water in channels was applied to the conditions of flow for the test pipe under consideration, for the purpose of ascertaining about what quantities of water might be expected. By consulting several authorities, 0.011 was selected as

the value of the coefficient of rugosity which seemed to be most generally used for wrought iron pipe. This value was used in the preliminary calculations and the result is represented graphically by curve *D*. Later, upon selecting a point from curve *B* and substituting in Kutter's formula, a value for the coefficient of roughness of 0.009125 was obtained. Then, choosing 0.009 for the coefficient, Kutter's formula was re-solved for various depths and curve *C* was obtained. The degree of conformity of the curves, *B*, and *C*, for depths less than 0.9 in. is very striking. For a full pipe, or for depths very near a full pipe, Kutter's formula does not conform to the experimental results. The reason for this is that just before a full pipe is reached the rate of change of the cross sectional area with respect to the wetted perimeter is rapidly decreasing. The hydraulic radius decreases as the area of a full pipe is approached and the effect in the formula is to decrease the quantity of water.

The Flow of Air

A number of tests were made to obtain the relation between the pressure drop and cubic feet of air per minute flowing through the dry pipe, Fig. 12. The curve shown in this paper was obtained with air entering the pipe at atmospheric pressure and at a temperature of 70 deg. fahr. Values shown by this curve are consistent with the charts for parallel and counter flow of water and air to be described in the following.

Parallel Flow of Water and Air

The principle achievement which may be claimed as a result of these experiments on the simultaneous flow of water and air through 1 in. pipe is represented by the parallel flow and counter flow capacity charts.

These charts show the relation between the pressure drop in ounces per 100 ft., the pounds of water per hour, and the cubic feet of air per minute. The manner of running the tests for obtaining the data for the chart for parallel flow was as follows:

The procedure was, first, to adjust the quantity of air. The valves in the air piping connected to the reservoirs were closed and the valves in the piping leading to the gasometer and to the atmosphere were opened. The vacuum pump was started, and a throttle valve located between the receiver and orifice meter was opened until the desired pressure was shown on the orifice manometer. As soon as the flow became constant, as indicated by the steadiness of the orifice manometer, the valve to the atmosphere was closed. The orifice manometer was maintained constant as the air was exhausted from the gasometer. The capacity of the gasometer was 5 cu. ft., and the time required to pass this amount was registered by means of a stop watch. Several trials were made and the cubic feet of air per minute for the given orifice manometer reading was determined. The valve at the outlet reservoir was then opened; the plug was removed from the tee connection at the inlet reservoir; and the valve connecting the gasometer to the apparatus was closed. The apparatus was then set for drawing air through the test pipe. The vacuum in the outlet reservoir was observed, and since the outlet reservoir was open to the atmosphere, this observation gave the pressure drop through the pipe corresponding to the existing rate of flow just determined. Everything was in readiness, now, for introducing the water. The valve in the line leading from the supply tank to the reservoir was adjusted, so that water flowed into the reservoir and through the test pipe at a depth of about one-tenth of one inch.

As soon as the flow became constant the readings of the following instruments were recorded: namely, the reservoir depth gage, the inlet depth gage, the weir

float gage, the hydrometric goblet, and the pressure drop manometer. The supply of water was then increased until the depth of the water in the reservoir had risen about 0.05 in. As soon as the flow had become constant the readings of the same instruments were recorded. The depth in the reservoir was increased about 0.05 in. again, and so on, until the test was terminated by the discontinuance of steady flow. During the entire test the rate of flow of air was maintained constant, and one such test as that described gave enough data to plot a constant volume line on the capacity chart. Similar tests were made for quantities of air ranging from about 1 cu. ft. per min., by $\frac{1}{2}$ cu. ft. intervals, up to about 9 cu. ft.

The mean depth of the water in the pipe is not shown on the capacity chart for the reason that the pipe depth gage readings did not give results which were considered as consistent and as accurate as the other observations from which the chart was obtained.

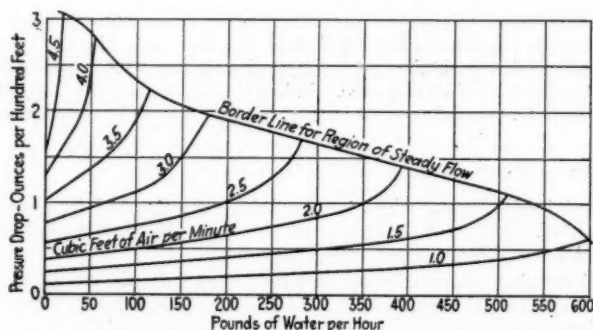


FIG. 14. A CAPACITY CHART FOR SIMULTANEOUS FLOW OF WATER AND AIR IN ONE INCH PIPE—COUNTER FLOW

Referring now to the capacity chart for parallel flow, Fig. 13, some of the characteristics of this condition may be studied. In the first place, there is a definite amount of air which may flow through the pipe for a given quantity of water and for a given pressure drop. Beginning with the vertical axis, the intersections of the constant volume lines with this axis give the pressure drops for zero water, or dry pipe, corresponding to the cubic feet of air per minute. As the cubic feet of air remains constant and as the quantity of water increases, the pressure drop also increases. This is due to the fact that the area for conveying the air has decreased and the velocity of the air has, therefore, increased. As the amount of the water is further increased a condition will finally exist where the flow no longer remains steady. At this point we have reached the critical capacity for steady flow. The border line on the chart for the region of steady flow was drawn through the final points on the constant volume lines which were obtained from the observations of the individual tests. The change from steady flow to turbulent flow is definite and very sudden. A phenomenon is present similar to that which is commonly attributed to water hammer in steam pipes, with the exception that in these experiments the vacuum is created in the outlet reservoir by means of a constant suction pump instead of condensing steam.

Counter Flow of Water and Air

For the counter flow tests there was no change in the method of making observations from that described for parallel flow. The only difference was in the manipulation of the valves to change the flow of air in the opposite direction, and in opening the outlet reservoir to the atmosphere. The plug in the tee connection at the inlet reservoir was inserted and the pressure drop manometer connection was changed so as to read the vacuum in the inlet reservoir.

The effect of counter flow on the capacity of the pipe is clearly shown by the chart, Fig. 14. For small rates of flow of water and air the differences between counter flow and parallel flow are also small. But for greater depths of water in the pipe and for higher pressure drops the critical capacities for counter flow occur much sooner than for parallel flow. This is, of course, as it should be, since the counter flow of air tends to impede the flow of water.

The effect on the depth of the water in the pipe in the case of parallel flow is to decrease the depth for a given amount of water as the quantity of air is increased. In other words, the air helps the water along. In the case of counter flow, the flow of air retards the flow of water, and, as the amount of air increases, progressively greater depths of water in the pipe are required for a constant rate of flow.

DISCUSSION

R. V. Frost: The practical value of Professor O'Bannon's work is in checking up the results of the Chezy formula which was used in making the calculations for the dry return pipe sizes in the last two editions of *THE GUIDE*.

The Chezy formula uses the pitch of the line as the principal factor and the tables as prepared give the slopes from vertical down to a pipe of 1 in. in 5 ft. and 1 in. in 10 ft., 1 in. in 20, and 1 in. in 30, and covers all sizes of pipe from $\frac{3}{4}$ to 6 in. Our difficulty at that time was to know exactly how to use that formula applying to iron pipe. We took that formula and applied it as far as we could, using the new factor 0.011 which Professor O'Bannon illustrated. Then we had the additional condition of the flow of air through the pipe to consider and this was applied on the theory of the usual steam formula. We had two conditions to take into account—the water and the air. We first were obliged to work out the tables for the amount of water that was flowing through the lines under different conditions. Then we had to calculate the amount of air, the maximum conditions, and see what the conditions would be under which the pipe was operating. We finally settled upon the condition that the pipe was running about three-sixteenths of its area with water and the balance with air. We took the condition of the full capacity of the radiation as 3 cu. ft. per 100 sq. ft., using that ratio. We determined that the air should be emitted from the radiator in the average time of about 10 min. On that basis the tables as they are given in *THE GUIDE* were worked out completely.

Now, to show you how closely our calculations check with what Professor O'Bannon has been able to determine, $\frac{3}{16}$ per cent of the area of the pipe falls somewhere in the neighborhood of the 100 line. We were about $\frac{9}{10}$ of an inch below that. We felt at the time we were working up these tables that a pipe would reach its maximum capacity at about 35 per cent. These per cents do not show on the chart as given, but Professor O'Bannon has them worked out on a chart that he has in his possession and that is what I am referring to. Thirty-five per cent falls on the

350 line. If you take the 400 line and take the air emission in 10 min. time, we will just about strike the line, the critical line. We will have an air emission of 4.8 cu. ft. per min.

Another conclusion that is to be drawn from the work of Professor O'Bannon is the effect of length of line on pressure drop. We have a pressure drop of about $\frac{1}{16}$ of an ounce for the normal conditions of condensation flow; so that condition can be increased, the length of line can be increased up to 500 or 600 ft. up to 1000 ft. before we would reach 1 ounce pressure drop, and I don't think that would ever come under practical conditions of working out a dry return line.

The experimental work of the Research Laboratory in Pittsburgh, given in the paper this afternoon, is another verification of that same idea, because they show that within the grades that we have normally up to 2 deg. all fall within practically a vertical line on their determinations.

J. A. DONNELLY: The formulas of Chezy, Weisbach, Unwin and D'Arcy all carry this square root of the mean hydraulic radius. For pipes running full the square root of the mean hydraulic radius is always one-half the diameter. That is common to almost all the formulas. It is further modified when we put in one plus something in the Kutter and one plus something else in the Unwin formula. Those were modifications of the Weisbach and Chezy. I don't agree with Mr. O'Bannon, but think if he checks up a little further he will find that the Kutter formula does correspond to pipes running full. That of all the formulas, due to the fact that as you cut down the wetted perimeter but a very small amount you have reduced the area very little, but the perimeter considerably, so that the maximum value of this for pipes running not full becomes as this percentage that he spoke of. But that is common to all formulas. The point arises then, immediately, even though this value is a little higher, except in very large pipes and very low flow, we would expect this wave motion because even though the air was standing still and the water moving we would have waves just the same as if the water were standing still and the air moving.

PROFESSOR O'BANNON: I might say that Kutter's formula is really a modification of Chezy's formula.

No. 690

AIR MOTION—HIGH TEMPERATURES AND VARIOUS HUMIDITIES—REACTION ON HUMAN BEINGS

By W. J. McCONNELL, M. D.¹ (NON-MEMBER), F. C. HOUGHTEN,² AND C. P. YAGLOGLOU (MEMBERS).

PITTSBURGH, PA.

THE proposed studies of the physiological effects of various temperatures and humidities on human subjects in still and in moving air while at rest and at work have been outlined in previous articles³ submitted to this

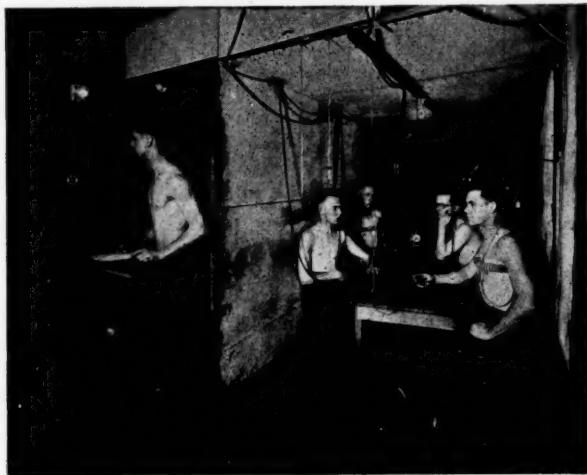


FIG. 1. WIND TUNNEL WITH SUBJECTS IN POSITION FOR TESTS

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⁴ The Effects of High Temperatures and Humidities, by W. J. McConnell, *The Nations' Health*, vol. 4, Oct. 1922, pp. 616-617. Further Study of Physiological Reactions, by W. J. McConnell, F. C. Houghten and F. M. Phillips, *Jour. AMER. Soc. HEAT. & VENT. ENG.*, Sept., 1923, [pp. 507-514].

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Society. Some physiological reactions to high temperatures and humidities in still air with the exposed subjects at rest were reported in a recent paper,⁵ and these may serve as a basis of comparison for other phases of the study. The results of experiments conducted under the same conditions of temperatures and humidities, but with different air velocities, are here presented with the view of illustrating the influence moving air exerts over still air.

In planning these later experiments we were confronted with the difficulty of obtaining a uniform air velocity over an area large enough to accommodate the subjects, and at the same time direct the current of air in such a manner that each subject shares its influence.

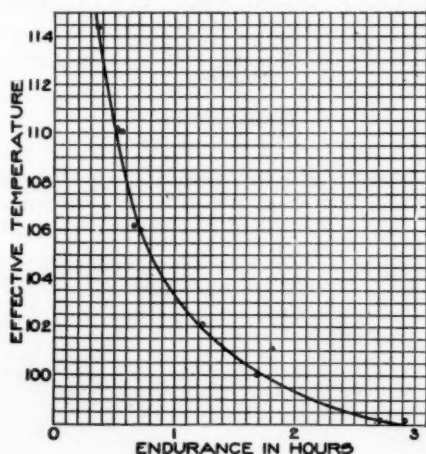


FIG. 2. AVERAGE TIME OF SUBJECTS IN TEST CHAMBER FOR TEMPERATURE ABOVE BODY TEMPERATURE

Air movement in mines and factories is usually incidental, and results from the system of ventilation employed. The workmen are exposed to these air currents from almost any angle, depending upon the position of the body. In sedentary occupations, where fans are used for cooling purposes, the individuals either face the source of air flow or receive it from a lateral angle. In these experiments, four of the subjects were seated as shown in Fig. 1, while the fifth, not shown, sat at the front end of the table facing the fans. The wind tunnel, which was constructed in one of the psychrometric rooms and designed to supply a constant flow of air, is also shown.

A detailed description of the tunnel, and the method of controlling and measuring the velocity of the air, accompanies the study of the effects of air motion on human comfort, which is presented in another paper.⁶ In this latter study the subjects faced the current of air.

The methods employed for collecting the various physical measurements were

⁵ Some Physiological Reactions for High Temperatures and Humidities, by W. J. McConnell and F. C. Houghten, *JOURNAL AMER. SOC. HEAT & VENT. ENG.*, March 1923, pp. 131-164.

⁶ Cooling Effect Produced by Air Velocities, by F. C. Houghten and C. P. Yaglou, *JOUR. AMER. SOC. HEAT & VENT. ENG.*

practically the same as in the still air experiments.⁷ One notable exception lies in the substitution of a mouth thermocouple for recording the oral temperature instead of the clinical thermometer used in the former experiments. The surface and rectal temperatures were taken by means of thermocouples, as described in the

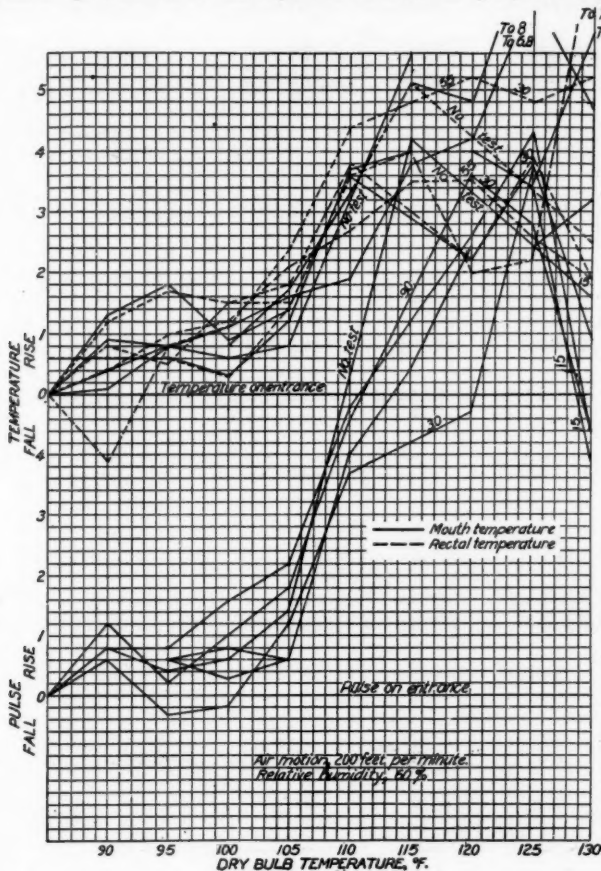


FIG. 3. INDIVIDUAL CHANGES IN TEMPERATURE AND PULSE RATE AT 60 PER CENT RELATIVE HUMIDITY AND 200 FT. VELOCITY OF AIR

previous study. The procedure followed in these tests under discussion remained unaltered.

Physiological Reactions in Still and Moving Air

Individual physiological reactions, as represented by the rise in body temperature, increase in pulse-rate, and loss in body weight, are given in Tables 1 and 2

⁷ W. J. McConnell and F. C. Houghten, work cited.

TABLE 1. RESULTS OF 200 FT. VELOCITY TESTS

Test No. and Date	Test Conditions	Subjects	Time		In chamber (hours)	Rectal Temperature		Pulse Rate in Beats per Min.		Rate of loss (lbs. per hr.)			
			Entered chamber	Left chamber		Initial	Time	Total rise per hour	Rate of increase	Initial	Loss		
55-A 10-1	D.B. 90.0 W.B. 67.4 R.H. 30%	Fulton McChesney Milliron Bishop Lynch	4:00	7:00	3:00	99.2 99.2 97.8 99.7	0.2 0.3 1.5 0.3	0.07 0.10 0.50 0.10	82 74 72 78	3:00	135.44 146.69 118.38 117.94 139.63	0.44 0.44 0.13 1.06 0.06	0.15 0.15 0.05 0.35 0.02
								Av. 0.19		Av. -1.0			Av. 0.14
56-A 10-3	D.B. 100.0 W.B. 74.3 R.H. 30%	McChesney Milliron Bishop Lynch	4:00	7:00	3:00	99.0 98.7 99.4	0.9 1.1 1.0	0.30 0.37 0.33	72 74 72 72	3:00	146.25 120.50 113.00 141.06	1.25 0.50 0.63 0.94	0.42 0.17 0.21 0.31
								Av. 0.33		Av. 4.7			Av. 0.28
57-A 10-5	D.B. 110.0 W.B. 81.7 R.H. 30%	Fulton McChesney Milliron Bishop Lynch	3:00	6:00	3:00	98.9 99.5 98.5 99.7	1.1 1.2 1.7 1.4	0.37 0.40 0.57 0.47	94 80 78 88	3:00	135.87 135.00 121.00 112.75 140.75	2.19 2.04 2.00 1.75 2.00	0.73 0.67 0.67 0.58 0.67
								Av. 0.45		Av. 2.9			Av. 0.69
58-A 10-8	D.B. 100.0 W.B. 99.9 R.H. 100%	Fulton McChesney Milliron Bishop Lynch	4:00	5:27 5:45 5:50 5:50 5:51	1:45 1:75 1:52 1:50 1:56	99.1 99.9 98.4 99.9	3.7 3.6 4.2 3.6	2.70 2.07 3.00 2.56	72 80 72 88	1:33 1:75 1:00 1:40 1:54	136.75 145.75 122.90 118.50 140.50	3.44 4.44 0.94 2.63 2.63	2.37 2.54 0.63 1.41 1.41
								Av. 2.58		Av. 32.9			Av. 1.72
59-A 10-10	D.B. 120.0 W.B. 89.1 R.H. 30%	McChesney Milliron Bishop	4:00	7:00	3:00	99.5 98.2	1.8 3.1	0.60 1.03	92 70 72	3:00	145.75 125.50 114.25	3.87 5.50 3.63	1.29 1.87 1.21
								Av. 0.82		Av. 9.1			Av. 1.46

10-A	D.B. 110.1 W.B. 110.4 R.H. 100%	Fulton McChesney Milliron Bishop	3:00	3:20 3:23 3:30 3:20	0:50 0:58 0:50 0:33	98.8 99.1 99.1 97.6	0:44 0:33 0:47 0:30	4.2 1.4 4.6 3.4	9.77 4.24 9.78 11.33	78 80 84 78	0:50 0:33 0:50 0:25	72 52 68 42	144.0 157.0 136.0 168.0	135.56 151.03 124.44 151.4	1.75 3.25 1.00 2.00	3.50 3.50 2.00 2.00	Av. 4.68	
61-A	D.B. 100.0 W.B. 87.1 R.H. 90%	Fulton McChesney Bishop Siegrifed	4:00	7:00	3:00	98.6 98.9 97.5 98.1	3:00	1.5 1.2 1.7 1.9	0.50 0.40 0.57 0.63	80 78 78 72	0:50 0:33 0:50 0:25	6 6 10 8	2.0 3.0 13.3 2.7	137.13 147.13 123.63 133.38	1.38 1.87 1.00 1.50	0.46 0.62 0.33 0.29	0.46 0.33 0.33 0.50	Av. 0.44
62-A	D.B. 110.1 W.B. 96.0 R.H. 60%	McChesney Milliron Bishop Siegrifed	4:00	7:00	3:00	99.3 97.9 98.0 98.8	3:00	2.7 4.4 3.8 3.6	0.90 1.47 1.27 1.20	76 76 70 70	0:50 0:33 0:50 0:25	48 48 38 40	16.0 16.0 12.7 13.3	147.50 124.25 116.00 133.50	5.87 2.25 3.63 2.81	1.96 0.75 1.21 0.94	1.96 0.75 1.21 0.94	Av. 1.2
64-A	D.B. 90.1 W.B. 90.0 R.H. 100%	Fulton McChesney Milliron Bishop Siegrifed	4:00	7:00	3:00	99.2 98.8 97.6 98.2 99.0	3:00	0.9 1.1 0.6 1.5 0.5	0.30 0.37 0.20 0.50 0.17	80 68 80 72 74	0:50 0:33 0:50 0:25	12 12 2 2 6	4.0 4.0 0.7 0.7 2.0	135.50 147.25 124.94 114.50 133.63	0.50 0.87 0.94 0.50 0.25	0.17 0.59 0.31 0.17 0.08	0.17 0.59 0.31 0.17 0.08	Av. 0.20
65-A	D.B. 129.3 W.B. 96.4 R.H. 31%	McChesney Milliron Bishop Siegrifed	4:00	7:00	3:00	99.0 98.2 97.8 98.9	3:00	5.1 4.0 4.6 3.7	1.70 2.15 3.86 2.01	72 78 70 78	0:50 0:33 0:50 0:25	72 58 58 72	24.0 38.7 29.0 39.1	147.19 123.56 115.50 130.00	7.50 2.69 3.13 1.43	2.50 1.45 1.43 1.43	2.50 1.45 1.43 1.43	Av. 1.79
66-A	D.B. 114.8 W.B. 100.0 R.H. 90%	Fulton McChesney Milliron Bishop Siegrifed	3:00	4:28 4:10 4:34 4:10	1:45 1:14 1:54 1:14	98.8 98.2 99.2 98.6	1:45 1:14 1:54 1:14	5.1 3.5 5.1 4.0	3.51 3.07 3.31 3.31	80 76 84 72	0:50 0:33 0:50 0:25	92 44 66 54	62.6 37.6 44.0 46.2	135.00 147.25 124.87 132.63	4.13 3.25 2.69 2.63	2.81 2.78 1.71 2.25	2.81 2.78 1.71 2.25	Av. 2.39
67-A	D.B. 90.1 W.B. 78.3 R.H. 60%	Fulton McChesney Milliron Bishop	4:00	7:00	3:00	98.6 98.3 98.2	3:00	0.8 1.2 1.2	0.27 0.40 0.40	72 76 70	0:50 0:33 0:50	8 4 6	2.7 1.3 2.0	134.50 148.50 125.00 115.00	0.44 0.50 0.19 0.50	0.15 0.17 0.06 0.14	0.15 0.17 0.06 0.14	Av. 0.14

TABLE 1. (Continued)

Test No. and Date	Test Conditions	Subjects	Entered chamber	Time In chamber (hours)	Rectal Temperature			Pulse Rate in Beats per Min.			Weight (Pounds)		Rate of loss (lbs. per hr.)
					Initial	Time	Total rise per hour	Initial	Time	Total Rate of increase	Initial	Loss	
68-A 10-31	D.B. 95.0 W.B. 85.3 R.H. 100%	McChesney Milliron Bishop Ferguson	4:00	3:00	99.2	3:00	1.6	70	3:00	26	148.63	3.63	1.21
					99.1	3:00	1.5	53		13	126.25	1.75	0.48
					98.9	3:00	1.5	73		14	126.25	1.75	0.48
					99.4		1.2	80		22	140.75	0.75	0.25
							Av. 0.48			Av. 7.1			Av. 0.62
69-A 11-2	W.B. 92.8 R.H. 30%	McChesney Milliron Bishop Ferguson	3:00	3:00	99.3	3:00	3.6	90	3:00	46	148.56	6.56	2.19
					98.6	3:00	1.20	90		56	127.94	4.81	1.00
					98.4	3:00	1.44	82		50	127.94	4.81	1.00
					99.4	3:00	2.8	84		68	137.50	3.50	1.10
							Av. 1.23			Av. 19.2			Av. 1.59
70-A 11-5	D.B. 95.0 R.H. 60%	Fulton McChesney Milliron Bishop Ferguson	4:00	3:00	99.0		0.5	80		4	135.00	1.13	0.33
					98.6		1.0	64		8	148.13	0.87	0.28
					98.1	3:00	0.8	82	3:00	2	129.50	1.06	0.39
					99.2		1.7	71		-3	116.38	0.75	0.25
							0.6	76		6	141.50	1.00	0.33
							Av. 0.31			Av. 1.1			Av. 0.32
71-A 11-7	D.B. 105.0 W.B. 105.0 R.H. 100%	McChesney Milliron Bishop Ferguson	4:00	3:00	99.0	0:48	3.8	78	0:47	72	146.63	2.63	3.76
					98.8	0:48	5.2	84		102	126.25	1.75	1.07
					98.6	0:50	2.5	69	0:50	71	116.00	1.94	3.88
					98.8	0:42	2.4	88	0:42	68	139.38	1.38	2.23
							Av. 4.71			Av. 120.1			Av. 2.96
72-A 11-9	D.B. 115.0 W.B. 85.3 R.H. 30%	Fulton McChesney Milliron Bishop Ferguson	3:00	3:00	98.6		1.7	90		6	133.87	2.00	0.67
					98.5		2.5	80		20	146.63	2.87	0.96
					98.0	3:00	1.6	72	3:00	18	127.06	2.56	0.80
					99.2		1.1	68		20	116.00	1.50	0.52
							Av. 0.63			Av. 4.8			Av. 0.80
73-A 11-14	D.B. 95.0 W.B. 70.8 R.H. 30%	McChesney Milliron Bishop Ferguson	4:00	3:00	98.8		1.4	68		10	146.87	1.25	0.42
					98.7	3:00	0.6	78		6	128.63	1.00	0.33
					98.4		0.9	72	3:00	5	116.38	1.38	0.46
					99.4		0.2	84		-4	141.06	1.31	0.40
							Av. 0.26			Av. 1.4			Av. 0.40
74-A 11-16	D.B. 105.0 W.B. 85.3 R.H. 30%	Fulton McChesney Milliron Bishop Ferguson	3:00	3:00	99.0		1.4	80		4	135.13	2.00	0.67
					98.6		1.5	80		8	146.13	0.75	0.25
					99.2	3:00	1.2	76	3:00	1	113.75	1.56	0.52
							Av. 0.48			Av. 2.1			Av. 0.53

TABLE 2. RESULTS OF 400 FT. VELOCITY TESTS

Test No. and Date	Test Conditions	Subjects	Time		Rectal Temperature		Pulse Rate in Beats per Min.		Rate of loss of (lbs. per hr.)	
			Entered chamber	In chamber (hours)	Initial	Time	Total rise (deg. rise per hour)	Initial	Time	Initial
2-16 1923	D.B. 105.0 W.B. 98.0 R.H. 10%	Herbert Fulton Boehm Guth Riley	11:00	3:00	97.7		0.0	72		141.64
					97.4		0.0	72		141.06
					97.4	3:00	1.9	62	3:00	167.19
					98.0		0.8	72		167.19
2A 2-19	D.B. 91.1 W.B. 60.2 R.H. 13%	Herbert Fulton Boehm Guth Riley	10:30	3:00	98.9		0.2	60		147.25
					98.9		0.2	60		147.25
					98.9		0.2	60		147.25
					98.9		0.2	60		147.25
3A 2-23	D.B. 105.9 W.B. 92.5 R.H. 90%	Herbert Fulton Boehm Guth Riley	10:30	3:00	99.1		-0.5	72		140.44
					98.7		-0.3	80		136.57
					99.0	3:00	-0.2	98	3:00	106.77
					99.2		-1.0	78		129.13
A 2-26	D.B. 100.0 W.B. 100.0 R.H. 100%	Herbert Fulton Boehm Guth Riley	10:30	3:00	99.2		1.6	66		141.75
					98.7		1.4	78		137.13
					98.7	3:00	2.2	72	3:00	166.94
					98.7		2.2	72		127.38
7A 3-2	D.B. 110.1 W.B. 98.0 R.H. 60%	Herbert Fulton Boehm Guth Riley	10:30	3:00	99.0		2.7	72		141.31
					99.0	1:00	2.7	72	1:00	141.31
					99.0	1:00	2.7	72	1:00	141.31
					99.0	1:00	2.7	72	1:00	141.31
9A 3-7	D.B. 120.0 W.B. 89.2 R.H. 30%	Herbert Fulton Boehm Guth Riley	10:30	3:00	99.2		1.5	80		140.00
					98.7		1.2	94		136.57
					99.0	3:00	1.6	78	3:00	129.19
					99.4		1.7	78		129.81

11A	D.B. 120.1	Herbert	12:30	0:50	95.0	0:50	3.7	7.40	72	0:50	78	156.0	140.69	2.19	4.30
3-12	W.B. 104.7	Boehm	12:00	0:60	99.0	0:57	3.2	5.62	72	0:60	28	46.7	164.00	2.00	3.33
	R.H. 90%	Guth	12:35	0:59	99.0	0:59	3.7	6.27	66	0:59	54	91.5	150.31	2.09	2.99
		Riley	1:00	1:00	99.2	0:92	5.1	5.54	66	1:00	102	102.0	128.38	Av. 3.47	
							Av. 6.21				Av. 99.1				
12A	D.B. 125.0	Herbert	1:30	2:50	99.0	2:45	2.9	1.18	68	2:50	52	20.8	140.61	4.73	1.89
3-16	W.B. 92.8	Fulton	11:00	3:00	98.4	3:00	2.6	0.88	72	3:00	54	18.0	138.81	5.06	1.99
	R.H. 30%	Boehm			99.0		2.9	0.97	66		54	18.0	104.27	4.61	1.54
		Guth			99.0		3.9	1.30	72		32	10.7	151.19	5.09	1.90
		Riley			99.0		3.5	1.17	66		66	22.0	127.44	4.75	1.58
							Av. 1.1				Av. 17.9			Av. 1.72	
13A	D.B. 115.4	Herbert	12:30	1:00	98.4	0:95	3.4	3.98	72	1:00	54	54.0	140.38	2.31	2.31
3-19	W.B. 99.9	Boehm	11:30	1:50	98.7	1:47	4.2	2.86	54	1:50	84	56.0	165.63	4.63	3.09
	R.H. 59%	Guth	12:30	1:00	99.0	1:00	3.1	3.10	66	1:00	38	38.0	151.00	2.38	2.38
		Riley	1:25	1:01	98.6	1:07	4.7	2.81	60	1:01	84	44.0	127.56	4.13	2.16
							Av. 3.09				Av. 48.0			Av. 2.49	
14A	D.B. 114.8	Herbert			98.4		0.9	0.30	72		12	4.0	140.77	2.11	0.70
3-23	W.B. 85.2	Fulton	10:30	3:00	98.8	3:00	1.1	0.37	80	3:00	10	3.3	140.75	3.06	1.02
	R.H. 30%	Boehm			98.8		1.2	0.40	66		24	8.0	164.00	3.22	1.07
		Guth			99.0		0.9	0.33	66		30	10.0	151.75	3.31	1.14
		Riley			99.4		0.7	0.23	66		30	10.0	128.81	3.31	1.10
							Av. 0.32				Av. 6.3			Av. 1.09	
15A	D.B. 95.4	Herbert			98.6		1.5	0.50	70		16	5.3	141.13	1.87	0.62
3-26	W.B. 94.8	Boehm	11:30	2:30	99.2	3:00	1.5	0.50	70	3:00	20	6.7	163.00	2.44	0.81
	R.H. 98%	Guth			99.0		1.0	0.33	66		12	4.0	148.25	2.00	0.67
		Riley			99.5		0.9	0.30	84		6	2.0	128.94	2.13	0.71
							Av. 0.41				Av. 4.5			Av. 0.70	
16A	D.B. 109.6	Herbert			98.5	0:43	3.0	6.97	72	0:25	60	240.0	141.31	1.69	3.93
3-20	W.B. 108.8	Fulton	11:00	0:42	98.5	0:42	3.7	8.02	80	0:40	70	175.0	140.44	1.63	3.88
	R.H. 98%	Boehm			99.0	0:43	3.2	7.45	72	0:48	83	173.0	165.00	2.00	4.17
		Guth			99.0		3.7	6.17	72	0:50	53	168.0	151.25	2.06	3.22
		Riley			98.3	0:60	3.7	6.17	72	0:62	117	188.7	127.38	2.06	3.22
							Av. 7.35				Av. 188.5			Av. 3.80	
17A	D.B. 120.0	Herbert			98.7	3:00	0.2	0.07	66		30	10.0	141.25	3.69	1.23
4-2	W.B. 78.0	Fulton	10:30	3:00	98.4	3:00	-0.1	-0.03	82	3:00	14	4.7	142.00	4.06	1.35
	R.H. 15%	Boehm			99.0	3:00	0.5	0.17	72		30	10.0	164.50	4.25	1.42
		Miller			99.6	1:00	0.1	0.10	84		36	12.0	130.87	2.53	0.84
							Av. 0.08				Av. 9.2			Av. 1.21	

TABLE 2. (Continued)

Test No. and Date	Test Conditions	Subjects	Entered chamber (hours)	Time Left chamber (hours)	Rectal Temperature		Pulse Rate in Beats per Min.	Total increase	Rate of increase	Weight (Pounds)		Rate of loss (lbs. per hr.)
					Initial	Total rise (deg. rise per hour)				Initial	Loss	
18A 4-4	D.B. 100.2 W.B. 74.7 R.H. 30%	Herbert Boehm Miller Siegfried	10:30	1:30	99.0	0.0	0.00	66	2	0.7	139.63	1.63
					98.7	0.9	0.30	66	6	2.0	162.50	1.80
					99.8	-0.5	-0.17	90	-6	-2.0	132.25	1.87
					99.4	-0.3	-0.10	72	0	0.0	130.50	1.63
						Av. 0.01	Av. 0.2	Av.	Av.			Av. 0.55
19A 4-9	D.B. 90.2 W.B. 90.2 R.H. 100%	Herbert Boehm Miller Siegfried	12:30	3:30	98.8	0.5	0.17	78	-6	-2.0	130.87	0.75
					99.0	0.3	0.10	72	0	0.0	163.91	0.78
					100.0	-1.0	-0.33	90	0	0.0	132.13	1.13
					99.4	0.0	0.00	84	0	0.0	132.75	0.38
						Av. 0.02	Av. -0.5	Av.	Av.			Av. 0.25
20A 4-13	D.B. 110.2 W.B. 81.8 R.H. 30%	Herbert Boehm Miller Siegfried	10:30	1:30	99.0	0.8	0.27	72	12	4.0	141.63	2.44
					98.4	0.7	0.23	88	8	2.7	139.44	2.50
					98.8	1.1	0.37	66	24	8.0	162.31	2.38
					99.6	0.1	0.03	98	4	1.3	130.50	5.00
					99.2	0.7	0.23	90	0	0.0	130.50	2.63
						Av. 0.23	Av. 3.2	Av.	Av.			Av. 1.00
22A 4-17	D.B. 90.6 W.B. 67.0 R.H. 28%	Herbert Boehm Miller Siegfried	10:30	1:30	98.4	-0.2	-0.07	70	-4	-1.3	141.38	0.94
					99.3	0.5	0.17	80	-8	-2.7	161.38	0.63
					99.0	-0.5	-0.17	96	-18	-6.0	130.94	1.75
					99.0	-0.1	-0.03	76	-4	-1.3	130.81	0.69
						Av. -0.03	Av. -2.5	Av.	Av.			Av. 0.33
23A 4-18	D.B. 90.0 W.B. 77.9 R.H. 59%	Herbert Fulton Boehm Miller Siegfried	10:30	1:30	98.6	0.1	0.03	72	0	0.0	141.31	0.50
					98.1	0.1	0.03	88	-12	-4.0	140.00	0.87
					99.1	0.0	0.00	72	6	2.0	163.50	0.56
					99.0	-0.1	-0.03	84	4	1.3	130.94	1.00
					99.0	-0.5	-0.17	80	-6	-2.0	130.63	0.13
						Av. -0.03	Av. -0.5	Av.	Av.			Av. 0.20
24A 4-20	D.B. 100.6 W.B. 86.9 R.H. 58%	Herbert Fulton Boehm Miller Siegfried	11:00	2:00	99.0	0.8	0.27	76	-4	-1.3	142.47	2.16
					98.3	0.7	0.23	88	-2	-0.7	140.56	1.87
					99.6	0.1	0.03	90	0	0.0	130.00	2.00
					99.2	0.2	0.07	78	10	3.3	130.38	0.67
						Av. 0.15	Av. 0.3	Av.	Av.			Av. 0.67
26A 4-25	D.B. 95.0 W.B. 71.0 R.H. 30.1%	Herbert Fulton Boehm Miller Siegfried	10:30	1:30	98.6	0.2	0.07	72	-6	-2.0	141.22	1.06
					98.3	-0.2	-0.07	84	-12	-4.0	140.50	0.87
					99.0	0.3	0.10	84	-12	-4.0	162.87	1.00
					99.2	-0.5	-0.17	84	-12	-4.0	130.50	1.44
					99.0	-0.4	-0.13	80	-12	-4.0	131.38	1.13
						Av. -0.04	Av. -3.6	Av.	Av.			Av. 0.37

27A	D.B. 95.1 W.B. 82.7 R.H. 60%	Herbert Fulton Boehm Miller Siegfried	10:30	3:00	99.0 98.5 99.1 99.0 99.6	3:00	0.4 0.7 0.4 0.2 -0.5	0.13 0.23 0.03 0.07 -0.17	72 84 84 86 86	-6 -12 -2 -6 -14	-2.0 -4.0 -0.6 -0.6 -4.7	140.81 140.56 140.56 140.56 131.38	1.10 1.25 1.25 1.00 1.00	0.40 0.42 0.33 0.33 0.33
							Av. 0.08			Av. -2.3				Av. 0.38
29A	D.B. 125.3 W.B. 108.3 R.H. 58%	Herbert Fulton Miller Siegfried	10:30	3:00	98.6 98.7 99.2 98.9	0:43 0:58 0:44 0:59	3.0 5.7 2.8 6.4	0.98 0.83 0.36 11.85	72 80 90 74	84 80 73 34	254.5 170.0 162.2 136.0	2.16 2.25 2.00 129.75	4.01 3.82 4.45 4.39	
							Av. 8.51			Av. 180.74				Av. 4.39
30A	D.B. 105.8 W.B. 78.8 R.H. 30.1%	Herbert McConnell Miller Siegfried	11:00	3:00	99.0 98.8 99.0 98.6	3:00	0.1 0.6 0.1 0.2	0.03 0.20 0.03 0.07	72 84 72 72	0 12 6 6	0.0 4.0 2.0 2.0	142.38 129.56 131.19 131.19	2.00 3.31 2.44 2.44	0.67 1.10 0.81 0.86
							Av. 0.08			Av. 2.0				Av. 0.86
31A	D.B. 130.0 W.B. 83.9 R.H. 15%	Herbert Miller Siegfried	10:30	3:00	99.0 98.6 99.0	3:00	1.4 1.9 1.4	0.47 0.63 0.47	72 90 84	18 30 18	6.0 10.0 6.0	142.00 129.38 131.00	4.44 5.06 4.50	1.48 1.69 1.50
							Av. 0.52			Av. 7.3				Av. 1.56
63A	D.B. 130.1 W.B. 113.5 R.H. 60%	Fulton McCasney Milliron Siegfried	3:00	3:00	99.1 99.1 98.6 98.8	0:25 0:25 0:27 0:39	2.1 1.9 5.2 5.9	8.49 7.60 19.26 15.13	84 82 82 70	36 41 44 82	211.7 164.0 176.0 205.0	135.00 148.19 127.06 134.00	2.00 1.06 1.50 1.63	6.45 3.03 4.28 4.08
							Av. 12.6			Av. 189.2				Av. 4.46
8-A	D.B. 106.0 W.B. 106.0 R.H. 100%	Herbert Boehm Guth Riley	11:00	3:00	98.6 98.9 99.0 99.0	0:54 0:66 0:62 0:84	3.1 4.1 3.8 4.0	5.74 6.21 6.13 4.77	72 60 66 72	72 72 54 78	122.0 107.3 87.2 92.8	141.69 165.28 150.00 129.87	1.06 1.78 1.81 1.81	1.80 2.66 2.15 2.20
							Av. 5.71			Av. 102.3				Av. 2.20
45A	D.B. 130.3 W.B. 96.3 R.H. 30%	Herbert McConnell Lincoln	12:30	3:00	98.6 98.8 98.8	2:00 2:00 1:58	3.0 2.6 3.9	1.50 1.30 2.47	72 78 78	66 56 44	33.0 28.0 31.0	140.28 140.13 151.87	4.41 4.25 4.00	2.21 2.12 2.13
							Av. 1.76			Av. 30.7				Av. 2.20

through the same range of temperature and humidity as employed in the still air experiments mentioned, with air velocities of 200 ft. and 400 ft. per min., respectively, and serve as a guide in comparing the various physiological responses in still and in moving air.

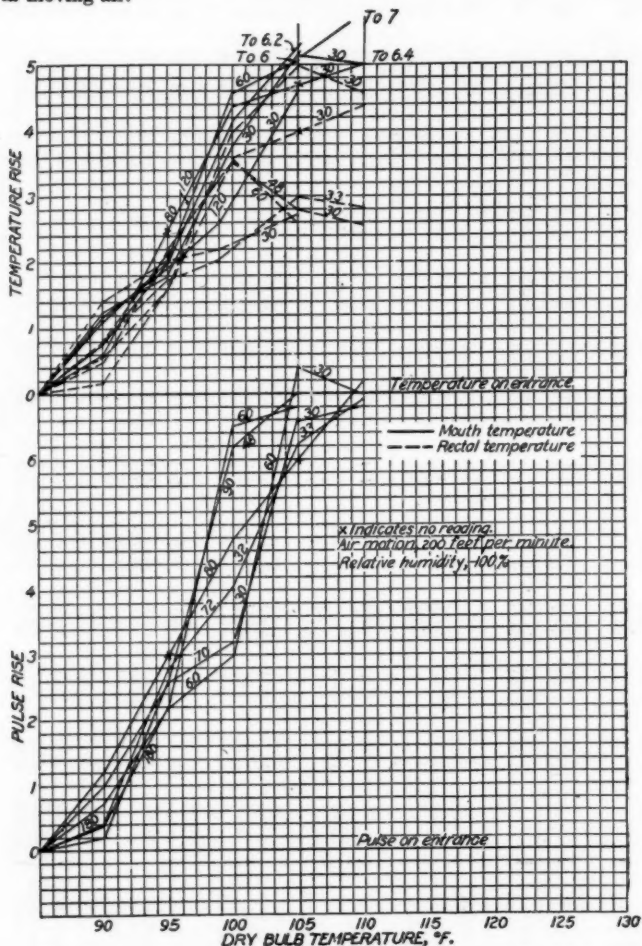


FIG. 4. INDIVIDUAL CHANGES IN TEMPERATURE AND PULSE RATE AT 100 PER CENT RELATIVE HUMIDITY AND 200 FT. VELOCITY OF AIR

Those wishing to make a minute study of the individual reactions for each subject may refer to these tables. The rate of reactions likewise is computed and then averaged, and columns 10, 14, and 17 represent the mean reactions of five subjects.

It has been pointed out that the experiments conducted in still air indicated that the upper limit of man's ability to compensate for atmospheric conditions lies around 90 deg. fahr. saturated (90 deg. effective temperature). Under similar conditions, but with an air velocity of 200 ft. per min., this limit is shifted to about 95 deg. as a result of the cooling effect of the wind.

Reference to the temperature and pulse curves plotted in Figs. 5 and 6 demonstrate the beneficial effects of air motion when the atmospheric temperature is below the body temperature. The converse is true, however, above body temperature, as shown by the still air reaction curves falling below those for moving air.

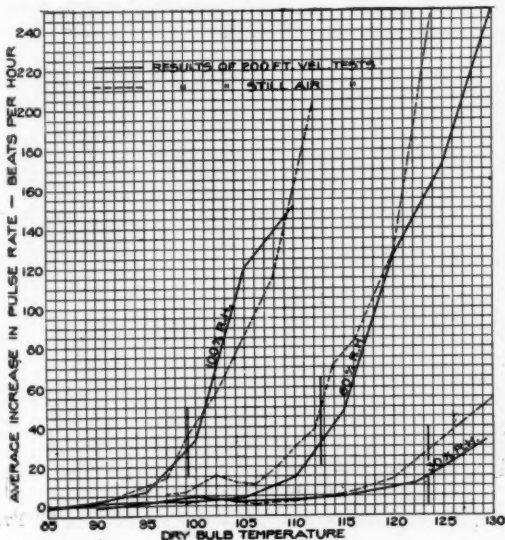


FIG. 5. AVERAGE INCREASE IN PULSE RATE IN THE 200 FT. VELOCITY TESTS

At 105 deg., with saturated air moving at a velocity of 200 ft. per min., the average rise in body temperature is approximately 1.0 deg. (or to be exact, 0.95 deg.) higher than is still air. The neutral conditions at which the effect of air motion ceases are shown by the three vertical lines for the different humidities. These were obtained in the experiments conducted to determine the cooling effect produced by air velocities.

The physiological reactions resulting from air moving at the rate of 400 ft. per min. are much more pronounced, especially at low temperatures. It is significant to note, however, that by doubling the velocity of the air, the physiological reactions do not double in rate of change. As stated above, the rise in body temperature was 0.95 deg. in air moving at 200 ft. per min. at a temperature of 105 deg. with a relative humidity of 100 per cent, while under the same conditions, but with the air moving at 400 ft. per min., the rise in body temperature is only 1.3 deg., the increase of 200 feet in velocity producing only 0.35 deg. additional rise in temperature.

Period of Endurance under Certain Atmospheric Conditions

The difficulties in determining the average period of endurance that subjects are able to tolerate certain atmospheric conditions are obvious. There is a strong disinclination to endure the discomfort, and individuals vary widely in their ability to remain in hot and humid atmospheres. It was observed that subjects remained exposed to adverse conditions longer after becoming accustomed to the routine of the experiments. In general, however, the fact that the subjects used in these experiments frequently had to be assisted out of the chamber indicates that they could not have remained a much longer period with safety.

The average time the subjects remained in the test chamber for various conditions in still and in moving air is shown in Table 3.

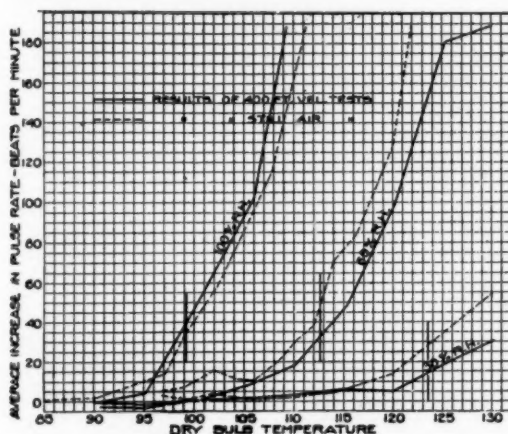


FIG. 6. AVERAGE INCREASE IN PULSE RATE IN THE 400 FT. VELOCITY TESTS

TABLE 3. TIME ENDURANCE IN TEMPERATURES ABOVE BODY TEMPERATURE (HOURS)
Period in Test Chamber

Effective temperature, deg. Fahr.	Test Conditions		Period in Test Chamber			
	Dry bulb, deg. Fahr.	Relative humidity, per cent	In still air	At 200 ft. velocity	At 400 ft. velocity	Mean time
98.2	110	60	2.22	3.00	2.91	2.71
98.2	125	30	2.92	2.94	2.90	2.92
100.0	100	100	1.69	1.60	1.79	1.69
101.1	130	30	1.36	2.22	1.86	1.81
102.1	115	60	1.00	1.35	1.35	1.23
106.0	106	100	0.78	0.68	0.68	0.71
106.2	120	60	0.69	0.63	0.67	0.66
110.0	110	100	0.75	0.43	0.49	0.56
110.2	125	60	..	0.50	0.52	0.51
114.4	130	60	0.39	0.38	0.35	0.37

A comparison of the different conditions does not show any appreciable change in endurance resulting from air motion. The mean of the results may serve as a fair index of the maximum period of endurance of individuals to high temperatures. These results are shown graphically in Fig. 2, where the average time in the test

chamber is plotted against effective temperatures. The hyperbolic curve fits the points remarkably well and indicates that the maximum endurance of the average individual exposed to an atmosphere equal to the body temperature is about $2\frac{1}{2}$ hours. The maximum endurance at 105 deg. effective temperature is approximately $\frac{3}{4}$ hour, and an effective temperature of 117 deg. limits the endurance to $\frac{1}{4}$ hour. By extrapolating the upper end of the curve, it is seen that it becomes asymptote to the temperature axis at an effective temperature of about 130 deg., at which condition the endurance of human beings is infinitely small. In other words, no human being could possibly exist in a saturated temperature of 135 deg. or its equivalent for any material length of time. The lower intersection of the

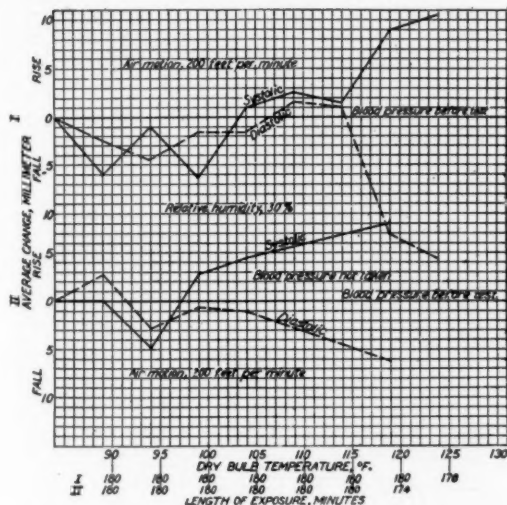


FIG. 7. AVERAGE RISE AND FALL OF BLOOD PRESSURE AFTER EXPOSURE

curve with the horizontal axis shows that a temperature of about 98 deg. can be endured for about 3 hours. The lower part of the curve cannot with any degree of certainty be extrapolated, due to the fact that the tests were never conducted beyond a 3-hour period. Furthermore, the cooling effect of the air would alter the course of the curve. It will be of interest to compare this curve, which represents length of endurance to high temperatures of subjects stripped to the waist at rest, with a curve plotted under similar atmospheric condition, but with the subjects at work, a phase of the study will soon be in progress.

Pulse and Temperature Changes

The importance of the circulatory system was discussed in a former paper,^{*} and the experimental evidence produced indicated that the pulse-rate was probably the best index to the severity of the discomfort due to high temperatures. The series of experiments just completed strengthen this view. Individual pulse and

* Work cited.

temperature changes are plotted in Figs. 3 and 4 for temperatures at 60 per cent and 100 per cent relative humidities, respectively.

These show quite a difference in the range of body temperatures for the individual subjects, while the increase in pulse-rate is much more uniform. The average increase in pulse-rate in moving air, compared with the rise in still air, is shown in Figs. 5 and 6.

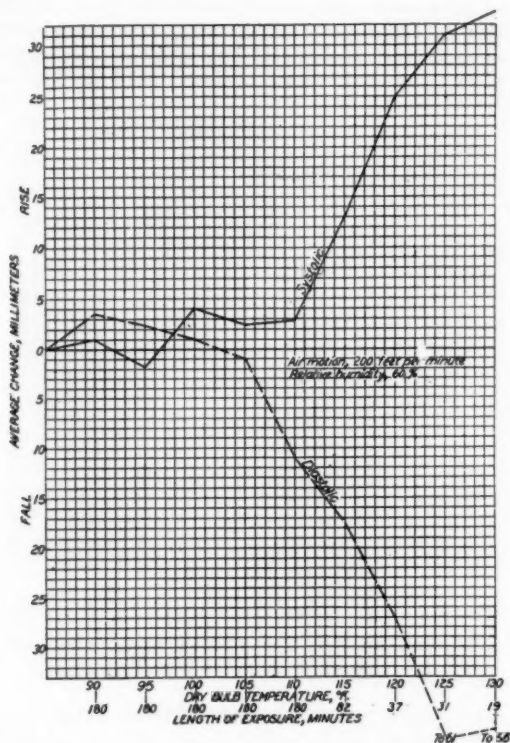


FIG. 8. AVERAGE RISE AND FALL OF BLOOD PRESSURE AFTER EXPOSURE

The average rise in pulse-rate of five subjects is plotted against dry-bulb temperature for three different relative humidities—namely, 100, 60, and 30 per cent. The still air conditions are shown by dotted lines. In moving air the increase in pulse-rate is slightly lower at saturation temperatures below body temperature, and considerably lower at 60 per cent relative humidity, than in still air of the same temperature and humidity.

Although a certain significance can be attributed to the rate of increase in body temperature, one must be guarded in using this rise as a criterion of discomfort. It is true that in some febrile diseases, and in an environment where the body tem-

perature approximates the air temperature, the pulse frequency correlates fairly well with the body temperature, as previously shown by Liebermeister,⁹ Mansfeld,¹⁰ Hill¹¹ and others. On the other hand, under the more extreme conditions of temperature, acceleration of the pulse exceeds the elevation of temperature in the majority of instances, and usually means some serious disturbance of the circulation. A high pulse-rate with a low temperature, associated with marked weakness,

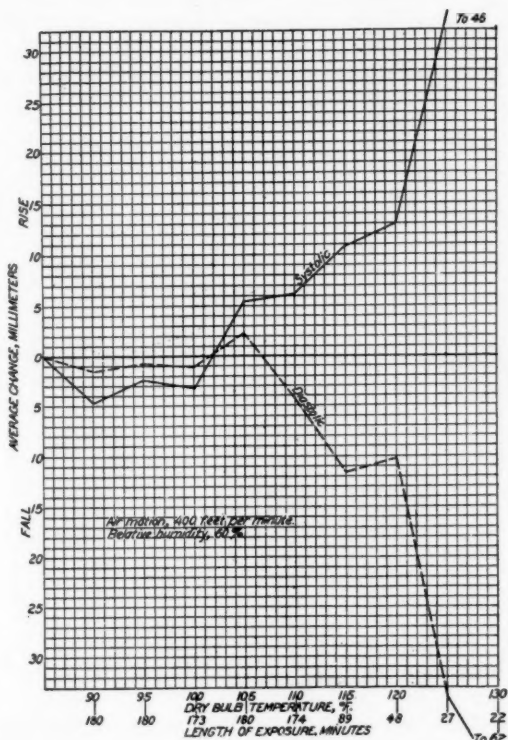


FIG. 9. AVERAGE RISE AND FALL OF BLOOD PRESSURE AFTER EXPOSURE

numbness, and profuse perspiration, is characteristic of the condition called "collapse." Subjects of these experiments have on occasions developed this condition. In one instance a subject after being removed from the chamber remained unconscious for 15 min., although his rectal temperature had not exceeded 101 deg. fahr. His pulse-rate, however, was very high, and was accompanied with a high systolic blood pressure and a minus zero diastolic pressure. In other tests this

⁹ Sahli's Diagnostic Methods, Potter; 2nd edition, 1911, p. 111.

¹⁰ Mansfeld, *Arch. Gesam. Physiol.*, vol. 134, 1910, p. 598.

¹¹ E. Vernon Hill's (Leonard) Medical Research Comm. Special Report No. 75, 1923.

same subject's rectal temperature exceeded 105 deg. fahr. without symptoms of collapse.

Blood Pressure

The increase in pulse-rate is accompanied by a marked dilatation of the peripheral blood vessels and an altered blood pressure. The systolic and diastolic curves follow an entirely different course as shown in Figs. 7, 8, 9, and 10.

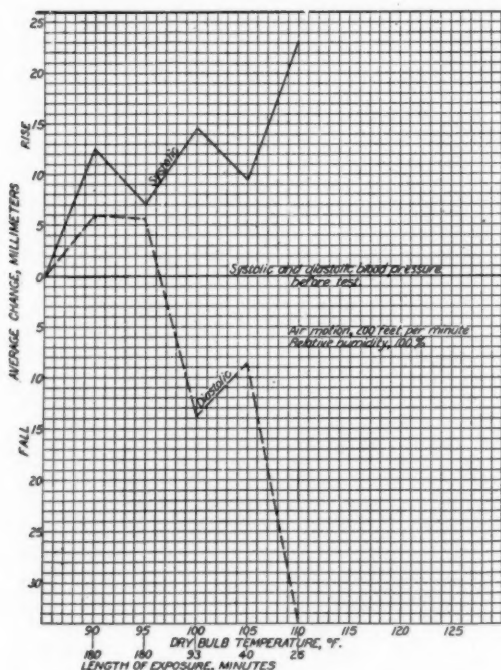


FIG. 10. AVERAGE RISE AND FALL OF BLOOD PRESSURE AFTER EXPOSURE

These show the average rise and fall in millimeters of mercury at the end of each test, represented by the solid and dotted lines, over the systolic and diastolic pressures before each test, indicated by the horizontal lines. These are plotted against dry-bulb temperature for three different relative humidities of 30, 60, and 100 per cent, with moving air at 200 and 400 ft. per min. Because of the few readings recorded at 100 per cent relative humidity with 400 ft. per min. air velocity the results are not plotted.

It may be recalled that in the still air experiments the pressures began to diverge at 120 deg. fahr., 30 per cent relative humidity; at 106 deg. fahr., 60 per cent relative humidity, and at 100 deg. fahr., 100 per cent relative humidity. In these experiments it is seen that these points are shifted slightly, due to the air motion.

In order to compare the rise and fall in blood pressure of a single subject with the average of five subjects, Figs. 11, 12, and 13 represent the measurements of a single subject.

Such a divergence of the curves seems very significant of a failing peripheral resistance, with a compensatory systolic pressure. At the end of an experiment in a high temperature the diastolic blood pressure was often found to be less than zero. This condition persisted only a short time after removal from the test chamber. The pulse-rate and mouth temperature likewise fell rapidly, while the rectal temperature persisted, and in some instance rose slightly for some time. With the exception of a slight headache and a tired feeling, the subjects always recovered rapidly.

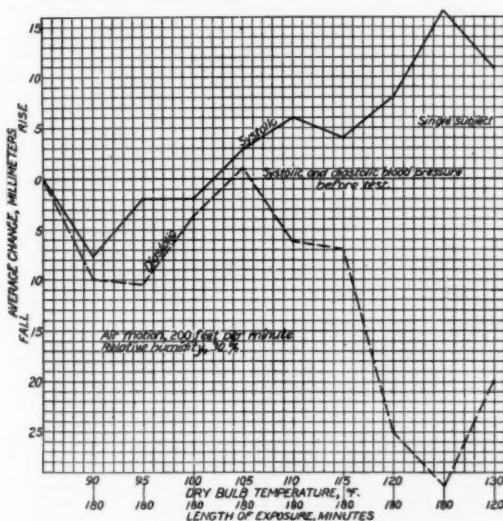


FIG. 11. RISE AND FALL OF BLOOD PRESSURE AFTER EXPOSURE

Body Temperatures

While oral, rectal, and surface temperatures were recorded, only the rise in the second is plotted against dry-bulb temperature for the various humidities. Surface temperatures will be the subject of a separate article. Figs. 14 and 15 give the average rise in body temperature (rectal) of five subjects plotted against dry-bulb temperature for three different relative humidities of 100, 60 and 30 per cent, at a velocity of 200 and 400 ft. per min., respectively.

The tendencies and general characteristics of the curves are similar to those for still air conditions, as shown by dotted lines. The effect of air motion is clearly exhibited by the 100 per cent and 60 per cent relative humidity plots, while at 30 per cent relative humidity it is not well pronounced, undoubtedly due to the fact that on account of the low humidity and slight effect that air motion might have, it is covered up by experimental error.

Figs. 16 and 17 show the variation in physiological reactions for the two air velocities. While at low temperatures the variation is measurable, at high temperatures no definite measure can be obtained due to the two curves crossing at several points.

It has been found¹² that low velocities are more efficient than high velocities, also that at temperatures above 90 deg., effective temperature variation in velocity of

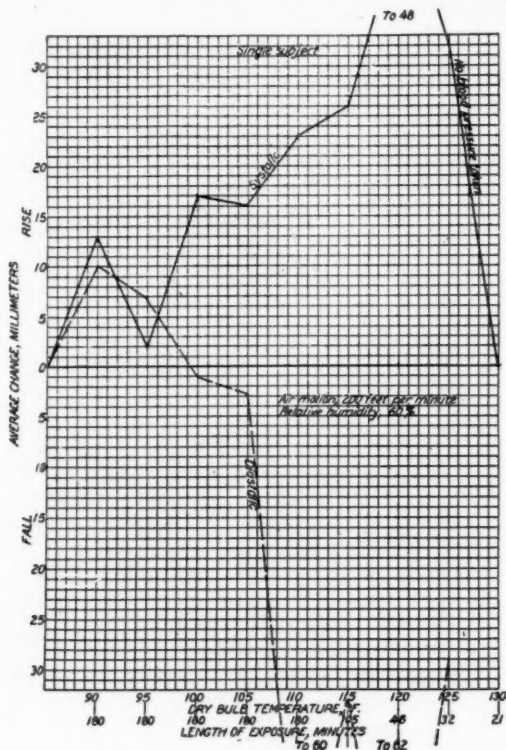


FIG. 12. RISE AND FALL OF BLOOD PRESSURE AFTER EXPOSURE

several hundred feet do not produce material change in the comfort of individuals. Unfortunately, up to the present time this work has been limited to the cooling zone only, so that a comparison cannot yet be made of the physiological reactions in still air and in moving air, on the basis of effective temperature. At the few low-temperature tests conducted, the results available indicate that the physiological reactions follow closely the scale of equivalent effective temperatures.

No attempt will be made to establish a measure of the changes in the body

¹² Work cited.

reactions due to air motion alone until the entire field of the investigation has been completed.

Effect on Respiration

In the experiments so far conducted—all subjects being at rest—the respiratory rate has not noticeably increased during the test, except in extreme conditions.

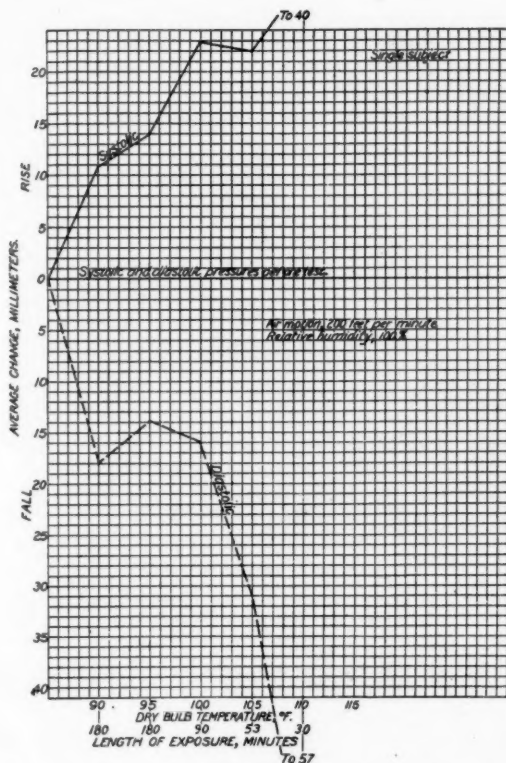


FIG. 13. RISE AND FALL OF BLOOD PRESSURE AFTER EXPOSURE

In the moderately warm experiments the respiratory rate frequently diminishes. In the extreme conditions the air is too hot to inhale through the nostrils and necessitates oral breathing. The subject apparently seeks relief through respiration, but on inhalation he finds no relief and immediately exhales so that the result is a series of short, irregular respirations. It is doubtful whether the ventilation of the lungs is increased under these conditions. In tests where the air is not too hot, and permits of nasal breathing, all subjects were able to hold their breath longer than in cooler atmospheres. In the extreme conditions, if the subject's attention is called

to his rapid breathing, and he is asked to breathe slower, he invariably does so. On the other hand, on leaving the test chamber, the respirations increase in depth and in number. A certain amount of relief is experienced in so doing, and the subject continues to breathe rapidly and deeply until he is relieved of the symptoms of discomfort he has just experienced.

Weight Loss

The average loss in weight per hour for the two different velocities is given in Figs. 18 and 19 plotted against dry-bulb temperature for the three different relative humidities.

It will be observed that the rate of change in the loss in weight is much smaller for low than for high temperatures, which is the natural result to expect. No direct

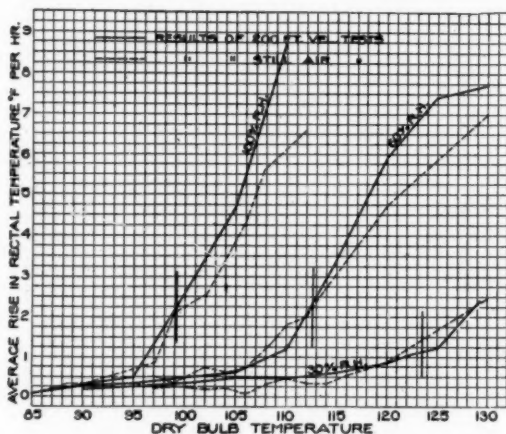


FIG. 14. AVERAGE RISE IN RECTAL TEMPERATURE IN THE 200 FT. VELOCITY TESTS

comparison can be made with the loss in weight for still air, due to the fact that the weights in the still air tests were not determined immediately before and after the tests, as they were in the last series of experiments.

That the series of experiments had no appreciable effect on the weight of the subject is apparent from the following Table 4.

TABLE 4. SHOWING EXPERIMENTS' EFFECT ON WEIGHT

When weighed	Subject					
	B.	M.	Fl.	F.	McC.	S.
Before series of tests, lb.....	117	118.6	135.7	140.12	146.11	133.6
After series of tests, lb.....	114	129.1	135.6	140.0	145.0	133.10

High Temperatures and Basal Metabolism

An extensive series of experiments has been conducted to determine the effects of high temperatures on the basal metabolism of the body and the results have been made a subject of another paper now in progress of writing.

Body Functions and Extreme Atmospheres

In conjunction with this work, considerable study has also been made of the regulation of body functions in extreme atmospheric conditions. Dr. Edward F. Adolph of University of Pittsburgh, who possessed data from an extended series of experiments upon 20 different men, continued his studies in the Research Laboratory of the Bureau of Mines and included a study of all data collected, together with the results of special experiments designed to determine the acid-base equilibrium, hydrogen-ion concentration, bicarbonate concentration, and the carbon dioxide dissociation curve of the blood; also, alveolar carbon dioxide tensions, the excretion of bicarbonate in urine, and the urinary hydrogen-ion concentration, and the changes

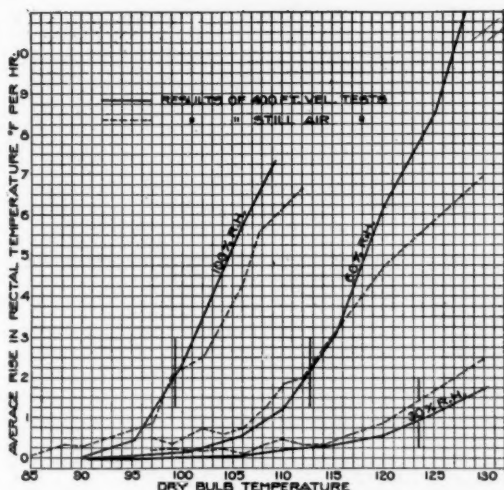


FIG. 15. AVERAGE RISE IN RECTAL TEMPERATURE IN THE 400 FT. VELOCITY TESTS

in the hydrogen-ion concentration of the sweat. This information is now available and is also in progress of publication.¹³

The urine was examined for albumin and sugar at intervals after exposure to the extreme condition, but in every instance the urine proved negative.

A record was kept of the food eaten by each subject; but upon careful study of the data collected no relation was revealed between the food eaten and the physiological reactions, with the exception that occasionally some article of diet not well prepared was accused of causing emesis during a test.

General Observations of Subjects

It is of interest to note that subjects who were compelled to leave a test on account of "feeling sick," although no marked change in their physiological reactions was recorded, invariably gave a history of constipation on that day.

¹³ The Effect of Exposure to High Temperatures upon the Circulation in Man, by Edward F. Adolph, assisted by William B. Fulton. To be published in the March, 1924, number of the *American Journal of Physiology*.

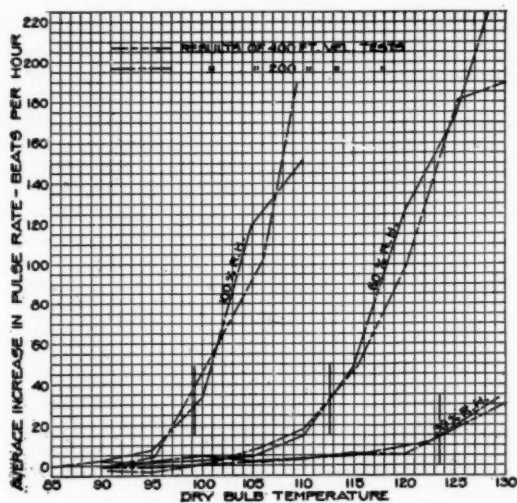


FIG. 16. VARIATION IN THE INCREASE OF PULSE RATE WITH VELOCITY

Certain individuals seem either to gain a rapid tolerance to high temperatures or are able to dispel the fear of danger from exposure. It was thought necessary to drop certain new subjects on account of their apparent inability to tolerate the conditions. Some would vomit shortly after entering the test chamber; others complained of the inability to breathe; while still others would not complain, but

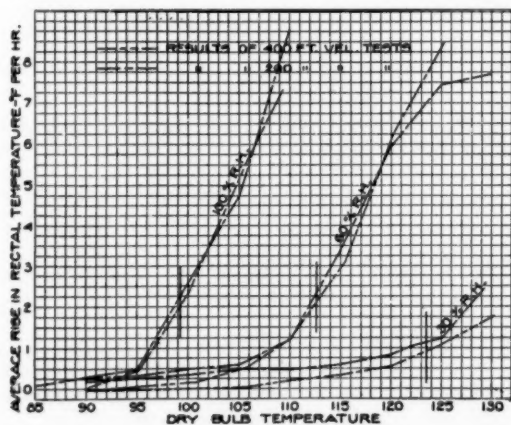


FIG. 17. VARIATION IN THE RISE OF RECTAL TEMPERATURE WITH VELOCITY

would simply faint. Later these same subjects proved to be among the best for endurance in the hot tests. No doubt fear of the unexpected was the influencing factor. A certain amount of fear was experienced by the best and oldest subjects on exposure to extreme conditions, as was noted in their unwillingness to remain in the test chamber alone.

The symptoms complained of on exposure to these extreme conditions of temperature are no different from those listed for the still air experiments, and include restlessness, irritability, headache, itchiness of the skin and scalp, palpitation, weakness, and subsequent fatigue.

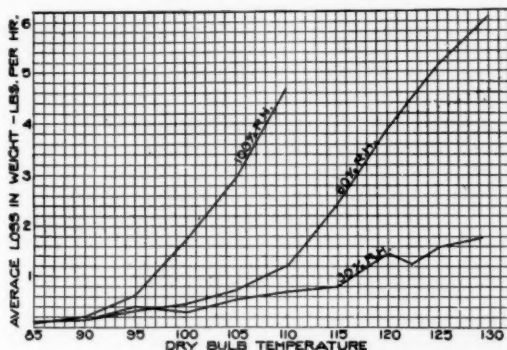
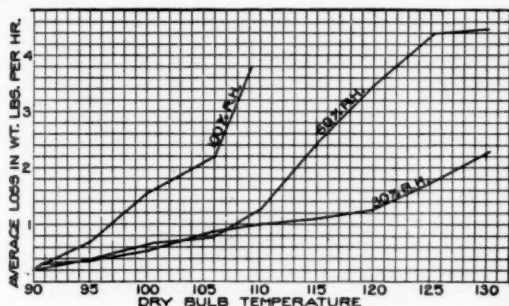


FIG. 18. AVERAGE LOSS IN BODY WEIGHT PER HOUR IN THE 200 Ft. VELOCITY TESTS

FIG. 19. AVERAGE LOSS IN BODY WEIGHT PER HOUR IN THE 400 Ft. VELOCITY TESTS



Conclusions

1. Air motion exerts a cooling effect on the human body in atmospheres where the temperature is less than that of the body; in temperatures above that of the body air motion increases the discomfort, but the rate of change in reactions cannot be doubled by doubling the velocity of the air.
2. Within the range of the experiments no appreciable change in the period of endurance resulted from air motion as indicated by the series of tests in still and in moving air.
3. The pulse-rate appears to be the best index to the severity of the discomfort.
4. The correlation between the pulse frequency and body temperature is not constant.

5. The systolic blood pressure increases on exposure to high temperatures while the diastolic pressure decreases, and frequently becomes a negative quantity.
6. The peripheral blood vessels dilate on exposure to high temperatures.
7. Respirations are increased in rate and depth after removal from a hot atmosphere to a cooler one.
8. The loss in weight which occurs after exposure to high temperatures is not permanent.
9. Exposure to high temperatures did not cause albuminuria in any of the subjects of these experiments.
10. Special studies of certain physiological reactions due to high temperatures are reported in separate articles.

Acknowledgments

Acknowledgment is here made to Dr. R. R. Sayers, chief surgeon, U. S. Bureau of Mines, under whose direction the physiological studies are conducted, for his many valuable suggestions and assistance since the study has begun; also to F. Paul Anderson, dean of the College of Engineering at the University of Kentucky, and director of the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS; and to A. C. Fieldner, superintendent and supervising chemist of the Pittsburgh Experiment Station of the Bureau of Mines, for their interest and encouragement at all times.

We likewise wish to record our thanks to those who have taken part in the experiments and assisted in the work. To W. Edward Miller, of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, for assistance in maintaining the temperature conditions, and to W. B. Fulton, R. E. Milliron, M. F. McChesney, T. G. Bishop, A. G. Lynch and G. W. Ferguson, students of the University of Pittsburgh and part time assistants of the Bureau of Mines, for their conscientious work in acting as subjects of the experiments, we are gratefully indebted.

No. 691

COOLING EFFECT ON HUMAN BEINGS PRODUCED BY VARIOUS AIR VELOCITIES

By F. C. HOUGHTEN,¹ AND C. P. YAGLOGLU,² PITTSBURGH, PA.

MEMBERS

THE fact that cooling may be produced by air motion dates from the earliest history of civilization. Egyptian kings were cooled by air motion produced by fans in the hands of their subjects.

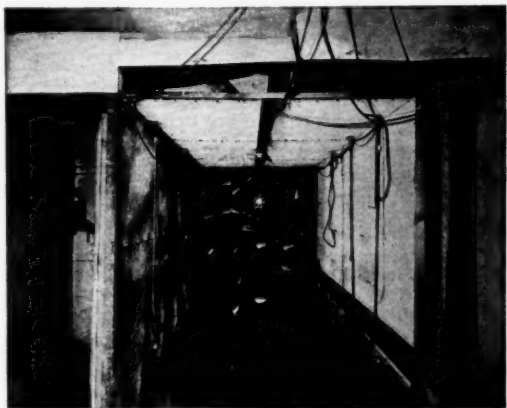


FIG. 1. WIND TUNNEL AND FANS IN VELOCITY ROOM

In the last decades of civilization, mechanical means have been devised for producing movement of the air and today practically all modern buildings are equipped with blowers and fans for ventilation and cooling purposes. While these developments a recentered on the importance of air motion in air conditioning very little attention has been given to establish a quantitative measure of the cooling

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produced on the human body by various air velocities of different temperature and humidity.

Dr. E. V. Hill has given the relation between wet-bulb temperature and cooling by air motion in connection with the Synthetic Air Chart. (TRANSACTIONS, A.S.H.&V.E., Vol. 26, 1920, page 540.) This relationship is in very close agreement with that given in this paper for a relative humidity of about 40 per cent. For higher and lower humidities, however, the relation does not hold true.

Harrington and Sayers of the U. S. Bureau of Mines have conducted an extensive investigation of the relation of temperature, humidity and air motion to human comfort (Physiological Effect of High Temperature and Humidity with and without Air Motion, by R. R. Sayers and D. Harrington, U. S. Bureau of Mines, Report of Investigation, Serial No. 2464). Their work was carried on largely

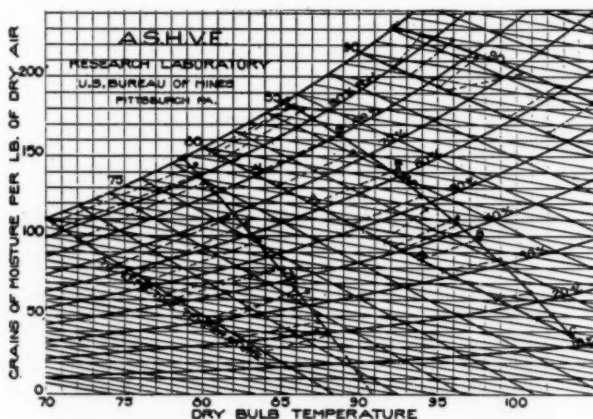


FIG. 2. PSYCHROMETRIC CHART SHOWING EXPERIMENTAL DATA OBTAINED ON THREE EFFECTIVE TEMPERATURE LINES AT 300 FT. VELOCITY

in hot metal mines of the west. While the reports give little or no quantitative data on cooling by air motion they are of great value in pointing out the relative importance of the three factors.

The cooling produced by air motion on the human body is necessarily governed by the difference in temperature between the body and its atmospheric environment. In order to maintain thermal equilibrium with the development of heat through metabolic processes within the body, the latter dissipates heat by radiation, convection and evaporation. Heat loss by radiation depends upon the character and temperature of the surfaces of the body and surrounding objects, and is independent of air motion excepting as the latter may vary these surface temperatures through a change in the rate of heat loss by evaporation and convection.

Convection in the strict sense of the word applies only to those air currents which are set up by a change in density of the air due to change in temperature. In the broader meaning of the word it includes air motion from any source. When used in the latter more general sense heat loss by convection is greatly affected by air

motion. The air in immediate contact with the body soon becomes heated to nearly body temperature as a result of which, heat is very slowly transmitted through the layer of warm air to the cooler air beyond. This so-called surface film of air plays a most important role in the transmission of heat from the surface of any body into the air. In the case of transfer of heat from a hot liquid through a metal pipe into the air, this surface film is the most important factor.

Air motion over the surface of the body tends to break up or decrease the effective thickness of the air film. The heated air near the surface which has lost its capacity for taking heat from the latter is removed and replaced by a fresh cool supply of air which in turn takes up heat to be removed by the moving stream of air. The greater the velocity of the air the more rapid this process and the greater the rate of heat loss by convection.

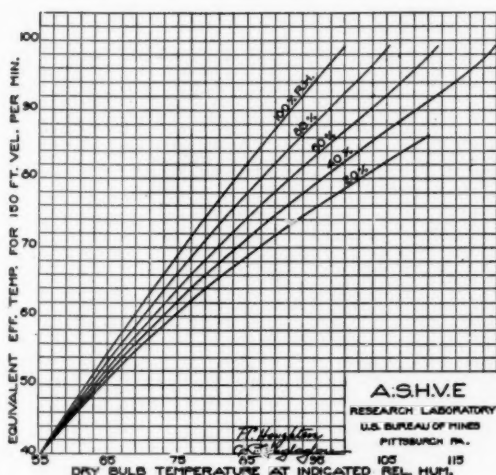


FIG. 3. RELATION BETWEEN DRY-BULB TEMPERATURE AND EQUIVALENT EFFECTIVE TEMPERATURE FOR 150 FT. VELOCITY

Air motion also plays a very important part in the loss of heat by evaporation. The same film of air next to the surface of the body tends to become saturated with water vapor, thus losing its capacity to evaporate water, as well as losing its capacity for taking up heat. Removing this saturated film of air and replacing it by air of greater evaporating capacity accelerates evaporation and hence heat loss by this means.

The Laboratory is making a study of the relation of temperature, humidity and air motion to human comfort. The work for still air has been completed and the results are published in two previous papers. (JOURNAL A.S.H.&V.E., March and September 1923.) Conditions giving the same feeling of warmth for various combinations of temperature and humidity within the limits of practical experience were determined and a scale of effective temperature for still air was developed. This scale is a true index of comfort.

This paper presents further information resulting from a continuation of the investigation with the additional factor of air motion.

Equipment

The data for this report was collected in the two psychrometric chambers equipped by the Research Laboratory at the Pittsburgh station of the U. S. Bureau of Mines. Two wind tunnels 5 x 7 x 10 ft. (large enough to accommodate two or three persons) of exactly the same type and construction were built, one in each chamber. To enable the walls and roof of the tunnels quickly to attain the temperature of the air, thin galvanized iron was selected for material. At one end of the

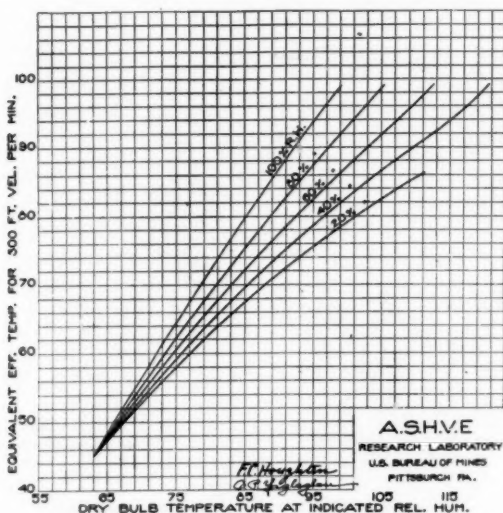


FIG. 4. RELATION BETWEEN DRY-BULB TEMPERATURE AND EQUIVALENT EFFECTIVE TEMPERATURE FOR 300 FT. VELOCITY

tunnel in one room referred to later as the velocity room, a bank of twenty 15-in. aero fans of the propeller blade type, was set up on iron frame work. The fans, with their guard frames removed, were arranged $13\frac{3}{4}$ in. from center to center in two vertical planes 15 in. apart, the tips of the blades overlapping about 1 in. To obtain a uniform velocity in the tunnel two eight-mesh screens were placed 1 foot from the blades of the fans and 2 in. apart.

The velocity of the air was varied by controlling the supply of air allowed to return to the fans through an arrangement of doors, around the front end of the tunnel. Small variations in velocity were accomplished by means of a rheostat in series with the main circuit.

Fig. 1 shows the tunnel with the bank of fans at the far end and the doors and shutters at the front. It was found that this arrangement produced a uniform

wind velocity over the cross-sectional area of the tunnel beyond $3\frac{1}{2}$ ft. from the fan blades.

Air velocities were measured by means of three anemometers. Two anemometers, one having a Bureau of Standards calibration were reserved as standard and only used occasionally to check the instrument in constant use. The standard instruments were never used in high humidities and handled with great care. They were occasionally checked with a Kata thermometer and other standard anemometers of the U. S. Bureau of Mines.

A preliminary survey disclosed that there was no measurable variation in velocity over the area of the tunnel beyond $3\frac{1}{2}$ ft. from the fan blades and a few inches from the floor and walls. All anemometer readings were taken with the instrument suspended at the level of the observers' shoulders near the center of the tunnel. Velocity observations were at all times made with observers standing at either side of instrument, so that the velocity recorded was that with the observers in the tunnel.

The judges stripped to the waist and wearing light-weight trousers stood side by side facing the fans at a distance of 4 ft. from the blades.

The tunnel in the still air room does not need any description as with the elimination of fan and shutter arrangement, it is exactly like the one described above.

Temperature readings in the two tunnels were obtained by means of automatic fan psychrometers properly screened from body radiation and checked frequently against sling psychrometer observations.

Method of Conducting Tests

Approximately 1000 tests were made with three judges in the majority of the tests, two of them taking part in all. A desired condition of temperature and humidity was produced in both rooms and maintained in the still air chamber. The fans in the velocity room were then started and the velocity adjusted to the desired value. Under these conditions the velocity room was considerably cooler than the other. The temperature and humidity of the former were then slowly raised keeping the relative humidity the same as that in the still air chamber, while the judges passed back and forth from one chamber to the other and recording each time individually the relative feeling of warmth of the two chambers. The temperature conditions of the rooms were recorded by one of the observers and kept unknown to the other two. As the temperature in the velocity room continued to rise a point was reached where the condition with air motion was equivalent to the still air condition and finally where the velocity room was the warmer of the two. The difference in effective temperature between the two rooms at the equivalent condition represented the cooling in degrees effective temperature produced by the particular velocity.

In a great number of tests the process was repeated in the reverse order to obtain a check on the first point. The velocity room was cooled gradually until the equivalent point was reached again and finally until the velocity chamber became the cooler of the two.

Table 1 gives the data collected in a representative test with 300 ft. air velocity. It will be seen that the change in effective temperature from the point where the observers agreed that the velocity room was cooler to the point where they agreed that it was warmer is only 1.2 deg. effective temperature. These conditions are shown by points A^1 and B^1 in the psychrometric chart, Fig. 2. By making other tests with the same effective temperature in the still air room but with different

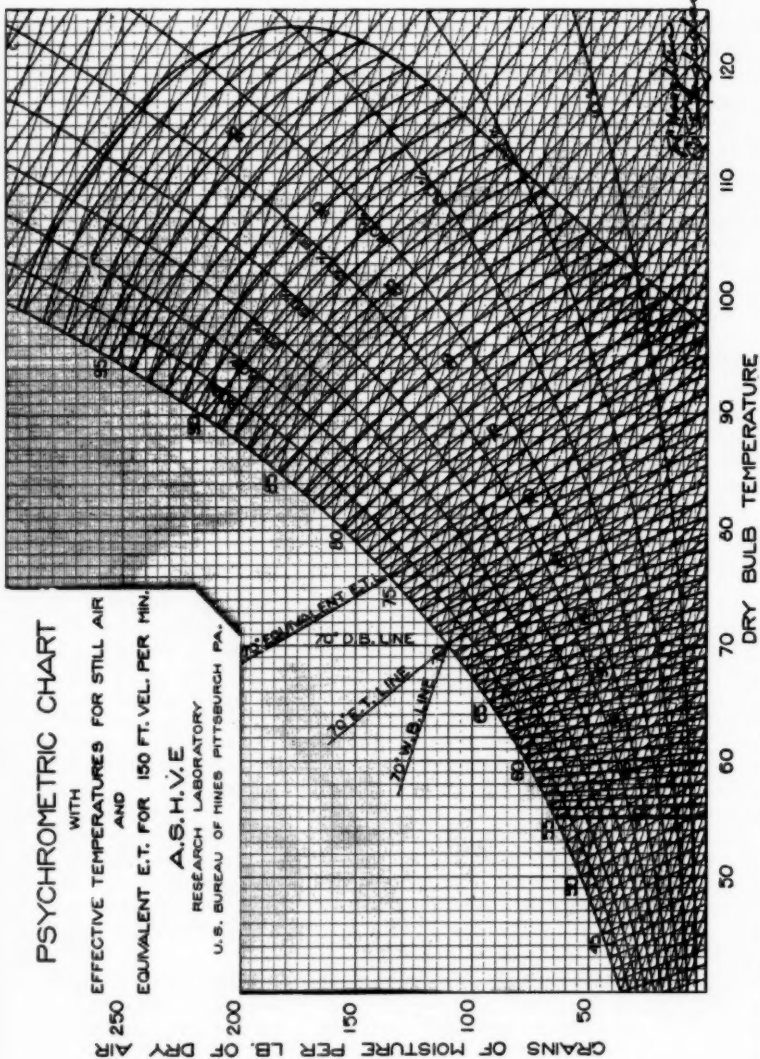


FIG. 6. PSYCHROMETRIC CHART WITH EFFECTIVE TEMPERATURE (FOR STILL AIR) AND EQUIVALENT EFFECTIVE TEMPERATURE FOR 150 FT. VELOCITY

relative humidities as shown by the various points, the corresponding equivalent conditions represented by points *B* were obtained for the same velocity. All points *B* give the same feeling of warmth with the given air velocity as points *A* in still air on the 80 deg. effective temperature line.

Since the line on which points *A* were located was called the 80 deg. effective temperature line for still air, in a similar manner the line averaging points *B* may be called the 80 deg. equivalent effective temperature line for 300 ft. velocity.

Data and Results

A large number of such lines were determined, ranging from 40 deg. effective temperature, the lowest attainable at the time of this writing, to body temperature and for various velocities from 150 ft. to 500 ft. per min. Three representative

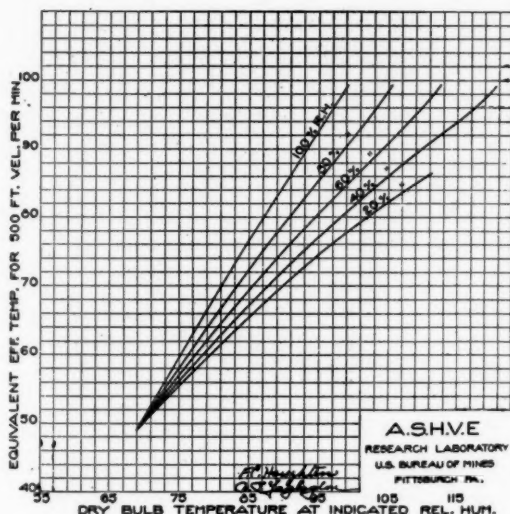


FIG. 5. RELATION BETWEEN DRY BULB TEMPERATURE AND EQUIVALENT EFFECTIVE TEMPERATURE FOR 500 FT. VELOCITY

lines are shown in Fig. 2 for 300 ft. velocity. The slope and curvature of the lines follow certain definite physical laws as shown in Figs. 3, 4 and 5, where equivalent

TABLE 1. SAMPLE OF TEST DATA TO DETERMINE EQUIVALENT CONDITIONS WITH 300 FT. AIR VELOCITY PER MINUTE

First Room Still Air		Second Room 300 Ft./Min.		A	Observers' Feelings Still Air Room Feels		
D. B.	W. B.	D. B.	W. B.		B	C	
83.5	78.6	85.9	81.8	Cooler	Cooler	Cooler	
83.4	78.5	87.4	82.7	Cooler	Sl. Cooler	Cooler	
83.2	78.5	88.0	82.9	Sl. Cooler	Sl. Cooler	Cooler	
83.3	78.4	88.7	83.2	Same	Same	Same	
83.4	78.5	89.0	83.5	Sl. Warmer	Same	Warmer	
83.3	78.7	89.3	83.9	Warmer	Sl. Warmer	Warmer	
83.5	78.7	90.3	84.6	Warmer	Warmer	Warmer	
83.7	78.8	91.2	84.9	Warmer	Warmer	Warmer	

The two equivalent conditions are shown by points *A*¹ and *B*¹ in Fig. 2.

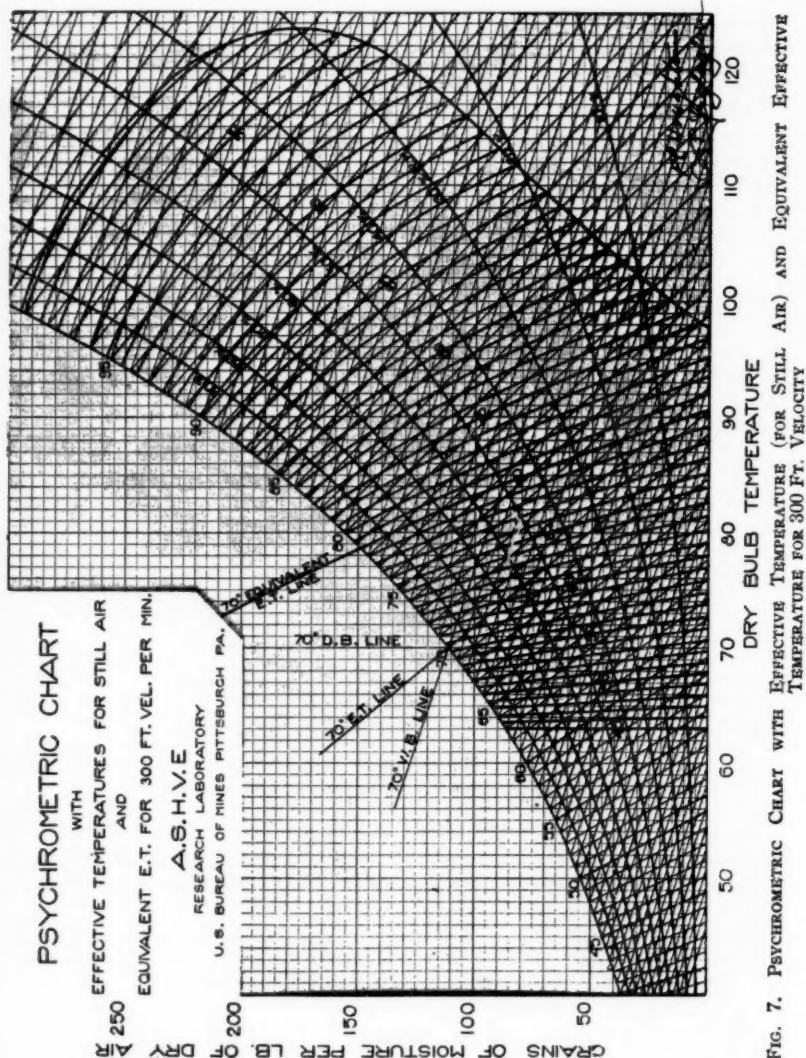


FIG. 7. PSYCHROMETRIC CHART WITH EFFECTIVE TEMPERATURE (FOR STILL AIR) AND EQUIVALENT EFFECTIVE TEMPERATURE FOR 300 FT. VELOCITY

effective temperature is plotted against dry bulb temperature at the points of their intersection with the various relative humidity lines.

It will be seen that all three sets of curves have the same characteristics and that the tendency of wind velocity is to swing the lower part of the curves to the right as the velocity increases. The effect of humidity is also shown very dis-

tinctly. At low temperatures each set of curves terminates at a common point where the effect of humidity is eliminated completely. In other words the equivalent effective temperature lines become parallel to the dry-bulb temperature lines. These three common points are on the 40, 45 and 49 deg. equivalent effective temperature lines for 150, 300 and 500 ft. velocities respectively. The upper ends of the curves are neutral points at which the effect of air motion vanishes completely, producing neither cooling nor heating on the human body.

The general characteristics of equivalent effective temperatures and their relation to still air effective temperatures are shown on the psychrometric charts in Figs. 6, 7 and 8 for the three different velocities, namely, 150, 300 and 500 ft. These charts are very valuable in so much as they show the interrelation of the information in Figs. 3, 4 and 5 to other physical qualities of air.

Attention is called to the fact that the equivalent effective temperature lines are not straight over the entire region investigated but form a series of curved lines parallel to the dry-bulb temperature at low temperatures and distinctly curved for temperature above the comfort zone. A comparison of the three charts discloses the effect of air motion on the slope and curvature of the lines. The higher the velocity the more pronounced the curvature of the lines.

The scale of equivalent effective temperature is given on the chart between 40 per cent and 50 per cent relative humidity. The numerical value of any line is fixed by the effective temperature to which it is equivalent.

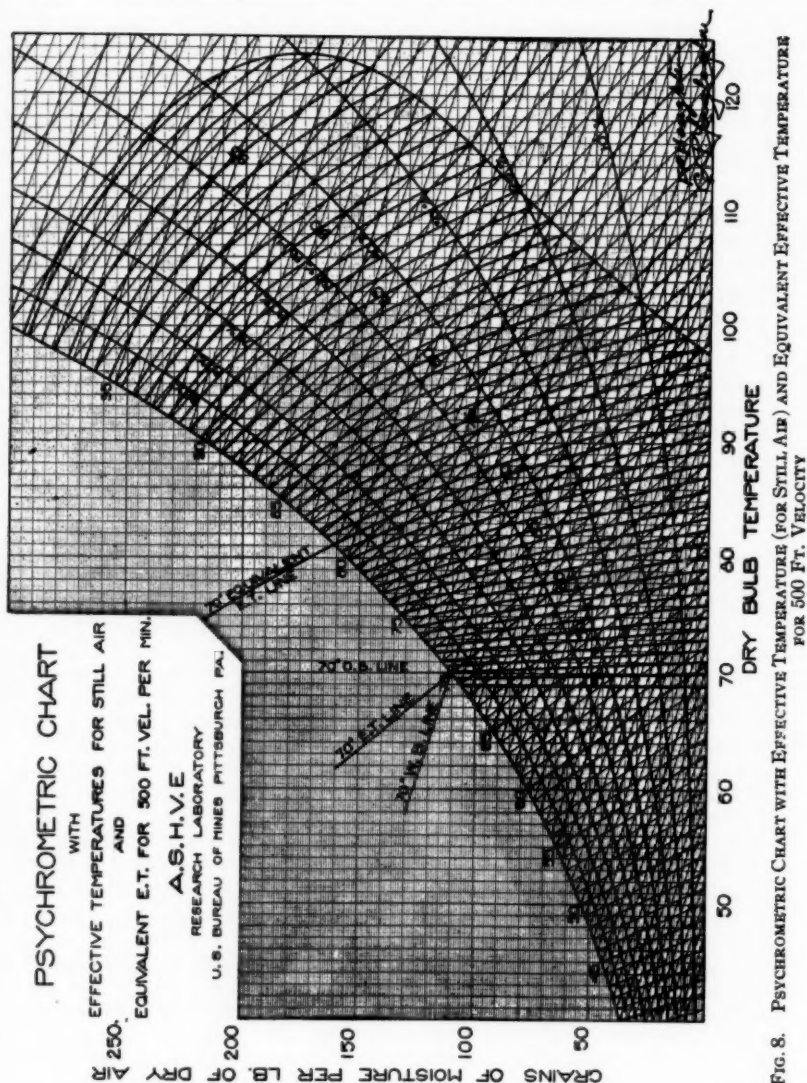
As mentioned above the cooling produced by a certain velocity is the difference between the effective temperature of the equivalent condition and that of the corresponding still air condition. It will therefore be observed in the charts that for the same velocity the cooling varies with body dry- and wet-bulb temperature.

As the difference in temperature between the body and its atmospheric environment decreases, cooling due to air motion also decreases and finally when the temperature of the environment reaches that of the body the cooling becomes zero. In other words there are certain definite conditions of temperature and humidity at which air motion is of no value whatever. These conditions may be called neutral, as far as air motion is concerned and a line joining them may be called the *neutral line*. Practically no variation in the neutral line has been found within the range of velocities employed in the tests and therefore they are drawn through the same points in all three charts.

The neutral line is a boundary separating conditions on the left where air motion produces cooling from conditions on the right where air motion produces heating.

The region of the chart so far investigated covers conditions between the neutral line and the equivalent effective temperature lines which are parallel to the dry bulb lines.

The charts in Figs. 9 to 13 give the cooling both in degrees dry bulb and degrees effective temperature for air velocities from zero to 700 ft. per min. They are for 20, 40, 60, 80 and 100 per cent relative humidity. The dry-bulb temperature of the various equivalent effective temperature lines at a given humidity as obtained from Figs. 6 to 8 is plotted against air velocity at intervals of $2\frac{1}{2}$ deg. The numerical value of the various curves at their intersection with the vertical axis, that is zero air velocity, gives the effective temperature for still air, for that particular relative humidity. Two parallel scales for dry-bulb and effective temperature at the left of the charts give the relation between these two temperatures for the particular humidity at zero velocity. The equivalent effective temperature scale is given vertically at the center of the chart.



In order to determine the equivalent effective temperature for any humidity and dry-bulb temperature follow the horizontal line through the given temperature to the air velocity and read the equivalent effective temperature on the scale of the curved lines. For example to find the equivalent effective temperature for 70

deg. dry bulb, 40 per cent relative humidity and 200 ft. air velocity, follow the 70 deg. dry-bulb line to 200 ft. velocity, on the 40 per cent relative humidity chart. This point is on the 56 deg. equivalent effective temperature curve. In order to determine the cooling in degrees dry bulb for this condition follow the same equivalent effective temperature line to the left to its intersection with the vertical axis at 61.0 deg. dry bulb. The difference between 70 deg. and 61.0 deg. or 9 deg. is the number of degrees dry-bulb cooling produced by the 200 ft. air velocity for this particular condition. The cooling in degrees effective temperature is given by the difference between 63.6 deg. E.T., and 56 deg. E.E.T., the effective temperatures for 70 and 61.0 deg. D.B. respectively found on the effective temperature scale opposite the dry-bulb scale. The cooling is 7.6 E.T. If the above information is desired for any condition whose relative humidity is not

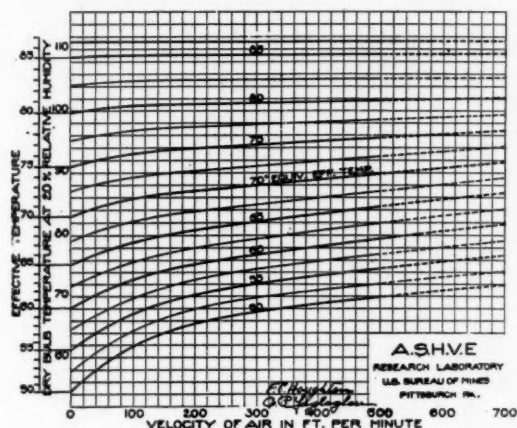


FIG. 9. CHART GIVING COOLING IN DEGREES DRY-BULB AND EFFECTIVE TEMPERATURE PRODUCED BY AIR VELOCITIES FROM ZERO TO 700 FT. FOR CONDITIONS WITH 20 PER CENT RELATIVE HUMIDITY

represented in the five charts it can be found for the same temperature but for the next higher and lower humidities and interpolating between these values.

Discussion

Several interesting facts are brought out by the psychrometric charts, Figs. 6 to 8 and the cooling charts Figs. 9 to 13.

1. The greater the air velocity the more nearly parallel the equivalent effective temperature lines become to the dry-bulb temperature lines. In other words as the air velocity increases dry-bulb temperature becomes more predominant as an index of comfort. Also the greater the velocity the higher the temperature, at which the equivalent effective temperature lines become vertical, in which case comfort is independent of wet-bulb temperature.

2. At ordinary temperatures the higher the relative humidity for a given dry-bulb temperature the greater the cooling produced by any air velocity.

3. There is no cooling produced by moving air for a dry-bulb temperature equal to body temperature at 100 per cent relative humidity and also for a slightly lower temperature at zero relative humidity. The highest temperature at which cooling results from air motion is 123.5 deg. dry bulb and about 30 per cent relative humidity.

4. For any condition there is greater cooling per unit increase in air velocity for low velocities than for high. Above 300 ft. cooling is approximately a

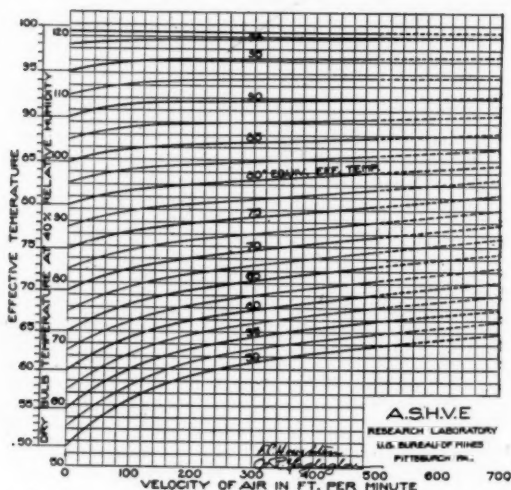


FIG. 10. CHART GIVING COOLING IN DEGREES DRY-BULB AND EFFECTIVE TEMPERATURE PRODUCED BY AIR VELOCITIES FROM ZERO TO 700 FT. PER MINUTE FOR CONDITIONS WITH 40 PER CENT RELATIVE HUMIDITY

straight line function of velocity. It is therefore evident that extrapolation beyond the limits of our experimental data gives results sufficiently accurate for practical purposes.

The above facts may be explained from a knowledge of the laws governing the production of heat in the body, its transfer to and dissipation from the surface. The heat developed in the body by metabolism is carried to a point near the surface by conduction and by the circulating blood. From this point it passes through the inactive part of the skin largely by conduction. The temperature of the body a very short distance within the surface maintains itself at a very uniform temperature of about 98.6 deg. Since heat passes through the outer skin by conduction there must be a temperature drop. That is, the true surface of the skin must be at a temperature somewhat below that of the body. After passing through the outer skin to the surface heat is lost by radiation, convection and evaporation.

Heat loss by radiation depends largely upon the difference in temperature between the surface of the body and surrounding objects. Loss by convection depends largely upon the difference between surface and air temperature. If the dry-bulb temperature of the air and surrounding objects is constant the only factor which can change the loss of heat from the surface of the body by either radiation or convection is the temperature of its surface. The human mechanism probably controls surface temperature slightly, and thereby heat loss by radiation and convection, by controlling the heat supplied for transmission through the inactive outer layer of the skin. Such a reduction in heat available for flow through the skin would result in lowering the skin temperature, not by increasing the temperature gradient through the skin itself but by increasing the depth of a smaller gradient resulting in a greater total difference between body and surface temperature.

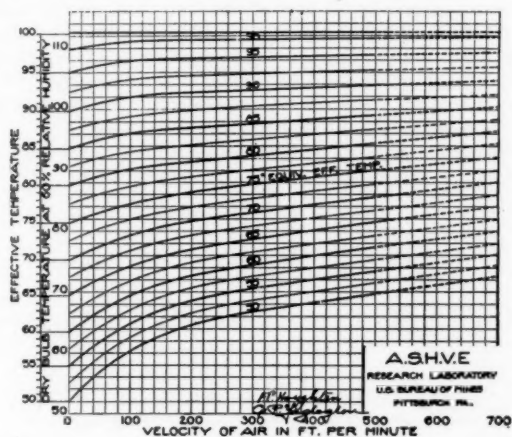


FIG. 11. CHART GIVING COOLING IN DEGREES DRY-BULB AND EFFECTIVE TEMPERATURE PRODUCED BY AIR VELOCITIES FROM ZERO TO 700 FT. FOR CONDITIONS WITH 60 PER CENT RELATIVE HUMIDITY

Heat loss from the surface by evaporation depends upon the amount of moisture available, the difference between its vapor pressure and the vapor pressure of the air, and air motion. Since the vapor pressure of the moisture on the surface depends upon its temperature the body cannot control this factor excepting as it controls the skin temperature. The difference in vapor pressure at any dry-bulb temperature depends, therefore, almost entirely upon the relative humidity of the air. The lower the relative humidity the greater the difference in vapor pressure and therefore the greater the evaporation. Probably the greatest control which the body has of heat loss is in the amount of moisture available for evaporation from the surface.

Let us consider the two relative humidities of 30 per cent and 90 per cent at 85 deg. dry bulb. At 30 per cent relative humidity and 85 deg. dry bulb the effective temperature is 73 deg. while at 90 per cent the effective temperature is 83 deg. As shown above the heat loss by radiation and convection for these two con-

ditions probably does not differ much. The heat loss by evaporation at the lower humidity would be considerably greater than that at the higher if the moisture available for evaporation from the skin was not limited. The amount available is, however, less than that demanded.

At the higher humidity there is more nearly as much as will evaporate than there is at the lower humidity. Air motion, therefore, will produce more cooling by evaporation at the higher humidity. This greater cooling at high humidities for any air velocity and temperature swings the equivalent effective temperature lines around so they become more nearly parallel to the dry bulb lines.

At body temperature and 100 per cent relative humidity, the body and skin attain air temperature and there is no change in heat loss due to air motion. Therefore heat loss by radiation, convection and evaporation ceases.

At a temperature slightly below that of the body with zero relative humidity the drop in temperature through the skin necessary for the transfer of the heat produced in the body, brings the skin and air to the same temperature resulting again in no heat loss by radiation or convection. The only loss in this case is by evaporation. The capacity of the air for evaporation is so great that all the available moisture is evaporated without the aid of air motion, hence it cannot be increased. This point could not be determined by direct experiments but the direction on the neutral line at 12 per cent relative humidity, the lowest point determined, and the intersection of the effective temperatures with the equivalent effective temperature lines located it with considerable accuracy at 97.7 deg. dry bulb.

Application

The information included in the psychrometric charts Figs. 6 to 8 and the cooling charts Figs. 9 to 13 finds many applications in the problems of the air conditioning engineer.

The resort to air motion as a means of cooling is as old as human experience.

For conditions where air motion gives considerable cooling it is probably the most effective and inexpensive means available. Unfortunately at high temperatures and humidities no great relief can be had by this means. The charts are valuable in showing what conditions can be improved or even what conditions will be made more unbearable by air motion. Whether it is desired to cool unbearable conditions in industrial establishments and mines or to make more comfortable conditions in theaters, churches and other public buildings the air conditioning engineer demands an answer to the following:

1. Will air motion improve the existing condition or make it more unbearable?
2. How much cooling in degrees effective temperature will a certain air velocity produce at a given temperature and humidity?
3. If the effective temperature of a given condition is too high for comfort or endurance and the cooling in degrees effective temperature necessary to alleviate the situation is known, what air velocity or what combination of air velocity and other changes will give the desired result?

The charts give all information necessary for answering the above questions. A glance at Figs. 6 to 8 will answer the first question. If the condition is to the left of the neutral line, air motion will improve it while if to the right, it will make the condition worse.

Given a condition of 90 deg. dry bulb and 60 per cent relative humidity (78.3 deg. wet bulb) what cooling will result from a velocity of 300 ft. per min? The chart, Fig. 7, gives the answer to this problem. The effective temperature of the condition is 82 deg. and its equivalent effective temperature 77.4 deg. The difference between the two or 4.6 deg. effective temperature is the desired cooling. If the velocity given was 350 ft. instead of 300, the cooling could be obtained from the chart, Fig. 11, which gives cooling at 60 per cent relative humidity for any air velocity from 0 to 700 ft. per min. The intersection of 90 deg. dry-bulb line with that for 350 ft. velocity gives an equivalent effective temperature of 76.8 deg. The difference between 82 deg. effective temperature found opposite the 90 deg. dry bulb on the effective temperature conversion scale, and 76.8 deg., gives 5.2 deg. effective temperature cooling. If the cooling was desired for 350 ft. velocity at 90 deg. dry bulb and 70 per cent relative humidity, it would be necessary to ob-

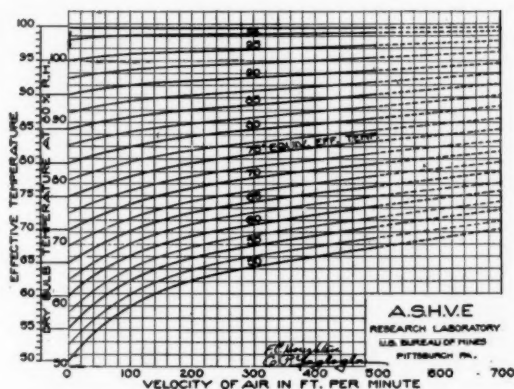


FIG. 12. CHART GIVING COOLING IN DEGREES DRY-BULB AND EFFECTIVE TEMPERATURE PRODUCED BY AIR VELOCITIES FROM ZERO TO 700 FT. FOR CONDITIONS WITH 80 PER CENT RELATIVE HUMIDITY

tain the cooling for 90 deg. dry bulb and 80 per cent relative humidity, from the chart, Fig. 12. The cooling for 90 deg. dry bulb and 80 per cent humidity is 5.3 deg. E.T. The cooling for 90 deg. and 70 per cent humidity would be approximately the mean of 5.2 deg. and 5.3 deg. or 5.25 deg.

The atmospheric condition in a certain place is 90 deg. D.B., 84.5 deg. W.B., or 86.1 deg. E.T., and it is desired to make it equivalent to 80 deg. E.T., by means of air motion. What velocity is necessary? Referring to the chart for 80 per cent relative humidity, Fig. 12 and locating 90.0 deg. D.B., on the scale of the latter follow the horizontal line through this point to its intersection with the 80 deg. equivalent effective temperature curve then down to the velocity scale where it is found that 420 ft. air velocity will give the desired result.

Another application of the cooling charts in conjunction with cooling by evaporation is worthy of consideration. A condition of 87 deg. wet bulb and 101.5 deg. dry bulb or 90.4 effective temperature and 56 per cent relative humidity existed

in the St. Johns Del Rey Mine¹ and it was desired to cool it to 80 deg. effective temperature. This condition was improved by cooling and dehumidifying, by refrigeration to 76.2 deg., wet bulb and 97.4 deg., dry bulb or 82.7 deg. effective temperature. We wish to determine from the charts if the same improvement could be obtained by an air motion or by evaporation and air motion.

The above condition (101.5 deg. dry bulb, 87 deg. wet bulb and 90 deg. effective temperature) may be made equivalent to 87.2, 86.2 or 85.2 deg. effective temperature with the aid of 300, 500 or 700 ft. air velocities, respectively. As determined from the charts this improvement is not equal to that actually obtained by refrigeration. If the air had been cooled by evaporation of water at the same temperature as the air, the wet-bulb temperature would have remained the same while the dry-bulb temperature would have dropped to the same value, or 87 deg. The effective temperature would also be reduced 87 deg. If the same air velocities

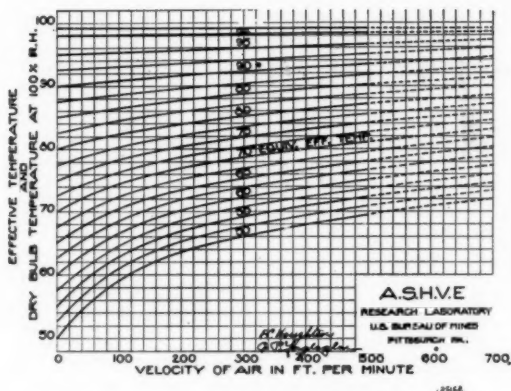


FIG. 13. CHART GIVING COOLING IN DEGREES DRY BULB AND EFFECTIVE TEMPERATURE PRODUCED BY AIR VELOCITIES FROM ZERO TO 700 FT. FOR CONDITIONS WITH 100 PER CENT RELATIVE HUMIDITY

are now produced, the condition of 87 deg. effective temperature will be made equivalent to 82.5, 79.9 and 77 deg. effective for the 300, 500 and 700 ft. velocities, respectively. Evaporation of water at the same temperature as the air and a velocity of 300 ft. would theoretically give the same result as obtained by refrigeration, while a velocity of 500 ft. would give about $2\frac{1}{2}$ deg. greater cooling. In practice the theoretical value cannot probably be attained. However, the cooling resulting from evaporation of water and 500 ft. velocity would probably be equivalent to that actually obtained at the expense of refrigeration.

It is of interest to note that for the original condition the cooling for the three velocities was 3.2, 4.2 and 5.2 deg., while for the saturated condition the cooling for the same velocities was increased to 4.5, 7.1 and 10 deg., respectively. This combination of cooling by evaporation and air motion is of great value in improving certain high temperatures where the relative humidity is not high.

¹ The Air Cooling Plant of the St. Johns Del Rey Mining Co., Ltd., Brazil. Trans. Inst. Min. Eng., Vol. LXIII, pp. 326-341 and 424-427, London, 1922.

Acknowledgments

The writers are indebted to W. H. Carrier and E. Vernon Hill for their interest in the work and the valuable suggestions offered. Acknowledgment is also due to W. E. Miller and R. L. Lincoln of the Research Laboratory, for serving as judges and the patience and faithfulness with which they endured the unbearable conditions associated with the tests.

DISCUSSION

W. H. CARRIER: I wish to point out some applications, some, that have already been made, and some that undoubtedly will be made in the future, and I want to show you something of their economic importance.

Mr. Houghten has already spoken along that line in discussing possibilities. I would like to give you some illustrations of accomplishment that parallel his predictions. All of these results, unless they are used, won't do us much good. There is very valuable information here if it is interpreted and its application shown; I think one of the things that we ought to do in the Society, either through committees or through the Research Laboratory, is a little education along the lines of application of this very valuable data.

Some time ago I had an illustration of a thing that Mr. Houghten very nicely pointed out, that in very high temperature conditions by saturating the air you could get an enormous cooling effect. I am going to show what that does on this chart, and then I am going to tell you where it is used and what it meant in dollars and cents. Take a condition of still air. We had 110 deg. and a dew point of about 80. This is in the rolling mill industry. Men tried to work in there. They had a lot of ventilation, even to give them that condition—120 deg., and radiant heat besides. But that would be on the comfort chart equivalent to about 87 deg. saturated. It is very near, you see, this borderline where a man would soon die, no bodily heat being removed. Men couldn't work for any length of time under those conditions. Of course the mills were shut down. By saturating the air and getting an air velocity we would get a comfort line that would correspond to 75 deg. still air saturated or very good working conditions for men that are used to heat.

Another illustration. In mines in India all the methods they had used of air blast, etc., on the men at certain conditions of weather, failed. They didn't have these charts then, but they took a chance and tried. They put on humidifiers and the results greatly exceeded their expectations, as you might see by the experimental effect, shown on this chart, of a draft of saturated air; no refrigeration. The air was simply cooled down by saturation and blown on the men. Where they were previously forced to shut down during long periods they were now able to continue with full effectiveness. That meant they could operate a period of time when other mills were shut down and during other periods when other mills were barely getting along. That was a chance discovery. With this data in our hands we can tell a man exactly what to do. The small result, of course, is to sell some apparatus. The big result is to increase production and it comes right back in dollars and cents, whether you live here in the United States or in India. The steel mills ought to pay money for work of this kind. One mill can afford, where they are operating under those conditions, to pay for this whole research out of their own pockets from the benefits they can get from it.

E. S. HALLETT: This is information that I have been looking for for a long time. I have been connected with designing auditoriums that have to be used in summer time and I have often wanted to know how much I could do, and I know now just what I can do. Nobody did know until these papers came out. Mr. Houghten told me yesterday what he was doing and I said, "That is the thing I have been wanting to know more than anything else." We had to travel in the dark. We knew there was something but not just what. As Mr. Carrier said, we don't have to guess about it now. We have the information and we can design to get cooling effect. It is a very great service to me personally.

E. VERNON HILL: There is one point in this connection that I would like to make and that is the application of this very valuable work. If we can't apply all this new data that the Research Laboratory is giving us, it doesn't fulfil its real purpose. One way, it seems to me, is to make the applications more understandable to the members of the Society. I presume that I represent the average intelligence of the members here and it is mighty difficult for me to follow these charts as they are prepared. We are opening up a new field; it is virgin in experimental work; and it seems to me that due consideration has not been given to the proper presentation of these charts. This data should be in a more readily understandable form.

Perhaps I am entirely wrong, but take for instance the equivalent temperature line running through 40 per cent relative and 68 deg. dry bulb. That would be somewhere around 64 deg. Now that is labeled 64 deg. equivalent temperature curve, presumably because it passes through the saturation curve at 64. Why wouldn't it be much simpler to call that the zero comfort line? We all know what the comfort line is. It means that any point on that line is comfortable and why call it the 64 deg. equivalent temperature line? Then, if we call that the comfort line, we could go up and speak of the unbearable conditions above that, and when you refer to those conditions on the chart it would carry a meaning to everyone here. It seems to me that the charts could all be simplified so that the members as a whole would get this thing much better than they do. I think if some attention was given to that the members would get this quicker without so much study and it would in this way be of much more value to them.

I do not think we should lose sight of the fact that when you are breaking new ground, going into new fields, you adopt names and chart forms, nomenclature and that sticks forever and a day.

F. D. MENSING: I certainly believe Doctor Hill is right. I hate some of the words we use. I think the word "psychrometric" should be wiped out. If anyone can give me a reason for it, I would like to hear it.

JOHN HOWATT: It seems that these curves were plotted from tests made on subjects stripped to the waist. I, like Mr. Hallett, am engaged in trying to design heating and ventilating plants for school buildings. Our subjects are not stripped to the waist. I wonder if it wouldn't be a good thing to have these curves put in some shape so we can use them. What is the effect of varying humidity temperature on people with ordinary clothing?

D. KNICKERRACKER BOYD: I feel that the valuable work shown by this research should be translated not alone for the benefit of the members of this Society, of which I am pleased to be one, but into a non-technical discussion for the man in the street. It seems to me that the work we have heard about tonight transcends the mere value of service to the engineering profession; it enters the realm of service to humanity; and I believe that the man in the street, the common everyday man

who is going to be benefited by this work, should be allowed to know more about it.

F. R. STILL: This investigation has just begun to scratch the surface of some of the things that we had in mind at the time we started this Research Laboratory. The paramount question that stands before us today is, what is it that the human system requires that makes for good ventilation, or an equivalent of what you get out in the country on a balmy spring day. We are just beginning to find out.

I was very much pleased in Detroit about a year ago, shortly after this work was submitted in Washington, to attend one of the moving picture houses one evening and I saw in the educational series they were running at that time pictures of just what is going on down there in that Laboratory.

E. B. LANGENBERG: There is one thing I have been missing, and that is the conclusions of our research staff on the tests that have been made. They left it for us to draw our own conclusions, and I would like, along with others, to have them set an average condition which we must meet out in the field in our actual installation work, so that we will have a basis to work on.

J. R. MCCOLL: These tests were all conducted with horizontal air currents, as I understand it. What would be the effect if the currents were from overhead or from mushroom systems below? That is one question. The other question is: We are assuming that the air velocities over a group of people are uniform for all of them. You seldom encounter that in actual practice. Which shall we take? Shall we take the average velocity for the whole crowd or take the maximum velocity?

H. W. BROOKS: There is considerable interest in this subject. In the past ten years it has been my misfortune to live under some of the conditions that were tested. I have lived in South America and Cuba, Santiago, Tampico and the City of Mexico, representative of a number of conditions which are given in these charts. There is one thing I want to point out which was covered by Mr. Carrier. The largest single item usually in industry is the labor item. A difference in the efficiency of labor of one per cent will more than offset a difference of ten per cent in any other department of the work. The biggest single factor in the hands of the efficiency or human engineer which has ever been published in all the story of efficiency engineering as we know it today is in the result of this paper.

F. C. HOUGHTEN: As Director Anderson said, we gave the question of terminology a great deal of consideration and considered a number of terms. There are several reasons for the words "effective temperature." We considered "sensible temperature" and a number of other terms but physiologists seem to rather direct us to the term "effective temperature." A number of physiologists of the Public Health Service have called it the "physiologically effective temperature" in view of the fact that it is the condition which produces physiological effects.

The question of the simplification of the charts came up. We had hoped to put this work in simpler form; it is not yet completed. We have not covered temperatures where heating results from air motion. Again, we want to do some work at lower temperatures. We were limited there by lack of cold weather. After getting the rest of the data we hope that we can put the charts in some simpler form.

The question of dress was brought up. We realize that the data is not in its final form until we have established just what effect dress has. A clothed person is not a sensitive instrument in determining coolness, as was brought out in the paper for still air read a year ago. We find that a person with few clothes on can determine a condition with a much greater sensitivity. For instance, if you are going

to compare conditions in this room with conditions in another room, with few clothes you can determine that to an accuracy of a fraction of a degree while with clothes the error is greater.

After doing this working on the subject with little clothing we intend to find the variation from our results with the subjects clothed.

The question was asked, as to whether the same effect would prevail if the wind direction were from overhead. We cannot say from direct experiment, although I believe it would have the same effect. We tried it from the back, front and sides, with no effect; we didn't think of turning our heads toward the wind.

Somebody brought the question of conclusions drawn from this data. I think the best form of conclusion is given in problems that we have shown solutions for. We mentioned a few solutions in the discussions, but there are a number of problems, very practical problems, worked out in the paper which I think answer the purpose of conclusions better than anything else.

No. 692

THE PLACE OF ELECTRICITY IN THE GENERAL HEATING FIELD

By LEE P. HYNES,¹ ALBANY, N. Y.

MEMBER

ELECTRICITY as a source of heat can no longer be ignored by the progressive heating engineer. No claim is made that it can replace coal, gas or oil fuels for general heating purposes, except under especially favorable conditions, but it has a logical place in the heating field and it is rapidly coming into its own.

The object of this paper is to point out the peculiar advantages of electric heat and to give the heating engineer practical data for readily determining when and how to use it in solving some of the especially perplexing problems which confront him.

Electricity is the most convenient form of heat, being instantly available through the touch of a button, and readily adjustable to the desired amount. It can be very accurately controlled by thermostats in large or small units, so that just the right amount of heat can be obtained at any desired location. This means efficient distribution.

It is surprisingly flexible in application and can be arranged to give any desired effect. For instance, it can be utilized to give direct radiant heat, combined radiant and convected heat, indirect convected heat with natural or power fan circulation, or it may be used as a heat source for circulating hot water or for the production of steam of any desired pressure. In other words, it is now entirely practicable to produce electrically any kind of heat effect that can be secured by any other means.

Electric heating requires no attendance, is clean, healthful, noiseless, and safer than other sources of heat. The *National Board of Fire Underwriter's* state on page 80 of their booklet of April, 1923, as follows: "It is recognized that electrical heaters are free from hazards peculiar to the use of fuel or gas heaters with their attendant hazards of the storage of fuel, ashes, the use of matches, flues, chimneys and in some cases open flames."

Unlike all other methods of heat production the conversion of electrical energy into units of heat occurs without loss. In other words, an efficiency in transfer of

¹ Consulting Engineer.

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100 per cent is secured. Furthermore this efficiency is inherent and is not reduced by neglect or unskilled management, as would be the case in the combustion of fuels.

In comparing efficiencies it is not necessary to consider that of the primary transfer of energy from fuel or water power into electric current or that of the distribution lines, because this is all included in establishing the rate per kilowatt hour at the user's meter. The user pays a fixed rate for energy used, and he gets all the heat paid for. There is no uncertainty about it. None goes up the chimney, there is no imperfect combustion, no wasted fuel, no overheated basement, or loss in distribution mains.

Furthermore there is less waste through overheating. The remarkable flexibility of distribution and control makes it possible to provide correct heating, electrically, with less total dissipation of heat units than with any other kind of heat. There being no vitiation of the air less ventilation is necessary than with many types of gas and oil heaters.

It can readily be conceded that electric heating is the ideal method where the cost is not too great. Under many conditions the cost is prohibitive but this is not always the case. Therefore it is essential correctly to analyze the question of relative costs before making a decision. To do this we must go further than simply to compare the thermal value per unit of cost of available fuels and electricity. In addition we must consider relative efficiency, interest on investment, maintenance, depreciation, and labor of attendance.

The efficiency of electric heating is 100 per cent while that of the average coal heater as usually operated is certainly not over 50 per cent, and that of gas and oil heaters 70 to 75 per cent. With fuel heaters the highest efficiency is obtained from large well proportioned units operated under favorable load conditions. In electric heating, however, the same high efficiency is secured in large or small units and under high, medium or small loads, and with either uniform or intermittent service.

In computing investment the chimney, extra roof expense, and a considerable part of the cost of the basement construction must be included for fuel plants. Maintenance must include all repairs to the entire heating system including chimney, fuel bins, coal windows, ash receptacles, as well as the furnace, boiler or radiators. Depreciation applies to all the above in varying percentages. The labor of attendance must include care of fires, removal of ashes, extra cleaning of basement and general premises. When all these legitimate factors of heating cost are properly considered, the expense for electric heating is frequently very little, if any greater, than that of other mediums and sometimes it is actually less.

There is so much variation in electrical rates in different locations that this is the first factor to consider. It is essential that rates similar to those allowed for power be secured for electric heating and most progressive power companies are now following this policy. Where current can be secured for 2¢ per K.W.-hr. it can be used for special applications, but where it can be obtained for 1¢ or less per K.W.-hr. it can be quite widely used. This is especially true where the climate is relatively mild, and the periods when heat is needed are of short duration or very intermittent. Also where good grades of fuel are expensive or hard to obtain.

In comparing fuel costs, the following data are useful:

1 K.W.-hr. of electricity = 3415 B.t.u. all effective.

1 lb. coal having 13,000 B.t.u. when used in a heating system at 50 per cent efficiency = 6500 B.t.u. effective.

1 cu. ft. city gas having 550 B.t.u. when used in a heating system at 75 per cent efficiency = 412 B.t.u. effective.

1 lb. oil having 18,500 B.t.u. when used in a heating system at 70 per cent efficiency = 12,950 B.t.u. effective.

The above fuels are equal in cost when the cost of 1 K.W.-hr. of electricity equals in cost 0.53 lb. of coal, 6.2 cu. ft. of gas, and 0.26 lb. of oil. Electricity at the rate of 1¢ per K.W.-hr. the above fuels would, to be equivalent, have to cost as follows:

Coal.....	\$37.73 per ton
Gas.....	\$ 1.61 per 1000 cu. ft.
Oil.....	\$ 0.28 per gal.

These figures represent the approximate relative costs for fuel only and do not include the many other factors which enter into the actual heating cost.

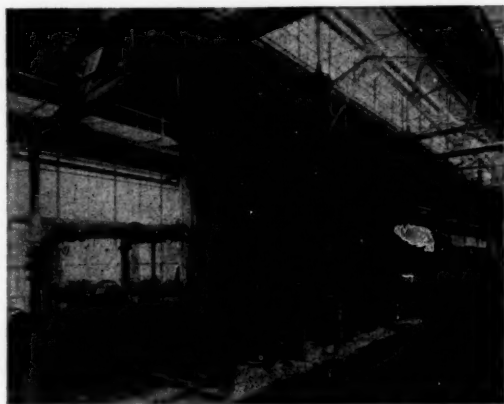


FIG. 1. TYPICAL EXAMPLE OF HOT WATER HEATING BY AN ELECTRIC BOILER

Even when electricity cannot be considered for the main heating it can be utilized to great advantage for special jobs. Frequently some additional heat is needed beyond the capacity of the boilers installed and the cheapest plan is to add electric units. Often some local heat is wanted at an awkward place to install piping, or heat is desired on short notice without having to start up the fires.

For remote places electric heaters are admirable. They are immune to freezing and are always ready for quick service. If connected with a thermostat they can be relied on to prevent freezing in pump houses, valve pits, and storage rooms, when a sudden drop in temperature occurs, and thus current is used only when actually needed. For temporary drying out of damp rooms or materials, portable radiators or hot blast heater units can be quickly installed at reasonable cost. Where there would be a fire risk from portable gas or oil heaters near inflammable materials, low temperature electric heaters may safely be used. Heaters can readily be designed to be entirely weather-proof and may even be buried in the ground to thaw snow and ice, or be submerged in water.

For many manufacturing and industrial processes electric heat is invaluable.

For japanning, or baking, the cost of current is a small factor when the resulting reductions in labor, time, and spoiled work are considered. New applications of electric heat to special machines are being constantly devised with great success and electric heat in industry is making great progress. Mild heat for candy making tables, or the intense heat in metal treating furnaces may be secured at will.

The heating engineer will readily see many applications where he can utilize electric heat to advantage, and the data which follows will aid him in selecting proper units for installation. The fact that electricity is to be used does not mean that the job belongs to an electrical contractor. It is fundamentally a heating job, and no one but an experienced heating engineer can successfully handle it, though, of course, the actual wiring must be done by an electrician the same as the wiring for a blower motor, or other unit of electrical apparatus with which the heating engineer has to deal.

As in all other heating problems the first thing to do is to compute the maximum B.t.u. required per hour, using exactly the same methods as for other forms of heat. As heating engineers have ample data for such computations no tables are given here. Then as 3415 B.t.u. equal 1 K.W.-hr. of electricity divide the total B.t.u. desired per hour by 3415, and the result is the total K.W. heater capacity required. As electric heaters are rated in kilowatts per hour, this total K.W. capacity is also the total maximum energy used per hour.

Type of Heater

After determining the necessary heater capacity the next step is to decide on the proper type, size, and location of the heater units. For general room heating there are three distinct types of electric heaters, radiant reflection, radiant convection and simple convection. Each type has its special uses, and the heating engineer must determine which to use in every case.

Radiant reflection heaters have refractory elements which are heated to a highly incandescent temperature. These are mounted in front of polished reflectors and practically all of the heat is thrown out in one direction as radiant and reflected heat. Heaters of this type in units of from 2 K.W. to 4 K.W. are ideal for installation in fire places or in similar recesses, where the problem is to get all the heat out into the room and prevent losses by conduction through the walls.

They are very effective for intermittent use in rooms not normally kept heated. Even if the temperature of the air in the room is quite low, the heat from this type of heater falling on the body of a person is absorbed and gives a pleasing warmth. Radiant heat is not effective for warming air, except as it is first absorbed by some solid body. Consequently it is the most efficient method of heating a solid body surrounded by cool air. Heaters of this type have rather frail units which may require occasional renewal. Portable heaters of this type of 600 watts capacity may be attached to lighting sockets but are too small for anything but very limited local warming.

The radiant convection type of heater is somewhat similar but more durable. Instead of having an incandescent refractory element this type has a cherry red heating coil of nickel chromium wire mounted on a substantial porcelain support secured to an insulated back plate. A considerable amount of radiant heat is thrown out by the coils and some heat is reflected from the porcelain supports. However, the heater also causes a circulation of air currents which carry away heat by convection and help effect a general heating of the air in the room.

This radiant convection type should be chosen for bath rooms and bed rooms

where local or quick heating is wanted. They are more comfortable in effect and much more durable than the more concentrated radiant reflection heaters previously described. They are made in varying sizes from 2 K.W. to 4 K.W. capacity and in two general styles. One is for flush or recessed mounting and the other for surface wall mounting.

The simple convection type heaters are the best for all around use, being similar in effect to steam or hot water radiators. They have substantial low temperature elements and should be used for air heating by either simple convection, or by forcing air over them by means of a fan. Heaters of this character can be stacked in standard units to secure any desired capacity. For ordinary room heating units varying from 2 K.W. to 4 K.W. are recommended.

In planning an installation of electric heaters it is important to know the approximate wall space required. For radiant reflection, radiant convection and simple convection heaters about 1 sq. ft. wall space per K.W. capacity are needed. This is about one-half the space per unit of heat that is required for a two-column steam radiator, hence in case of limitation of room space electric heaters have the advantage of occupying less space for equal heat than steam radiators, and, of course, they have a still greater advantage over hot-water radiators in this respect. Heaters for surface wall mounting will usually extend out from the wall about the same distance as a two-column steam radiator.

Another interesting fact is that an electric heater designed for a desired capacity will always deliver it. The output of heat units is not dependent on the rapidity of air circulation or the finish of the outer surface. This differs from a steam radiator which simply holds back the heat, keeping it as steam until it can condense. An electric heater is entirely different in principle, as it must deliver its full heat under all conditions. If the outer surface is coated with aluminum paint or some other poor radiating material, the surface temperature will simply rise until the heat is given off as rapidly as generated. In an electric heater a good radiating surface and a good design of air circulation mean lower radiator temperatures, not less heat given out.

Control Systems

Thermostatic control should always be used for convection heaters and can be arranged in groups of heaters in any desired manner. Rooms opening together may be controlled as a unit if desired, but independent rooms should have independent control. In very large rooms the heaters can often be divided among two or more thermostats and local temperature maintained as desired, at less total expenditure of energy, than is possible with one unit of control.

Even for small installations thermostatic control is available, as no air or other separate medium of energy is required. The electric current is the source of power for the control as well as for the heat. For small units up to 1 K.W. direct-acting thermostats of suitable design may be used, but for larger capacities the thermostat should simply control a relay switch.

Hand control of individual heaters can sometimes be dispensed with, but in large rooms it is often advisable to have certain heaters installed with hand control switches unless more than one thermostat is used as described above. In any case each group of heaters should have a master control switch by which the heat may be entirely shut off, or the thermostat put in control or removed from control as desired.

Thermostatic control is not needed for radiant reflection and radiant convection

heaters, as they should be hand controlled, as desired. Being for localized heating, thermostatic control would not be satisfactory. Thermostats are sensitive to air temperatures, while these radiant types of heaters are designed to give a cheerful sense of warmth to the body rather than to heat the air.

Flexibility Features

Frequently an invalid or elderly person requires one or two rooms warmer than the general household require. This is especially troublesome in the changeable spring and fall seasons, or when the fire gets low or a sudden cold snap occurs. Electric heaters are an excellent solution of such problems. By installing ample capacity in convection heaters under thermostatic control, no heat will be wasted. The regular heating system can function normally, and if a few more degrees of heat are needed the electric heaters supply it. This automatic topping off of local heating is very effective, as even a few degrees variation mean much for the comfort of anyone who is inactive or feeble, and the cost for electricity is relatively small.

For large storage or work rooms in stores and factories, it is often advisable to use one or more unit heaters with power fans. These give excellent distribution of heat and are very easily installed. They should be controlled by thermostats and should be designed so that in case the fan is stopped the current is cut off from the heater coils to prevent overheating them. A blower unit of this kind is very compact and requires no attendance. It can be suspended from a wall or overhead to save floor space and needs no chimney or fuel storage.

Hot Water and Steam Systems

We have seen how well adapted electricity is for warming air and will now consider its use for producing hot water and steam. For domestic hot-water supply, a re-circulation type of electric heating drum and a storage tank is more practical. The heater drum should be piped up the same as a gas heater and a good grade of insulation should be used to cover the drum, piping and storage tank. At least 1 in. of the best quality insulation should be used, as otherwise a great loss of heat will occur. Thermostatic control should always be used for economy.

In considering the installation of an electric water heater it is essential to know what capacity is needed. This depends not on the size of storage tank but on the amount of water drawn in a specified time. Flow heaters are totally inadequate except for very small demands. The only feasible plan is to provide ample storage in a well-insulated tank. This is at once evident when we check up the heat units required for water heating.

One B.t.u. will raise 1 lb. of water 1 deg. fahr., and as 1 gal. of water weighs $8\frac{1}{2}$ lb. it will take 8333 B.t.u. to raise 10 gal. of water 100 deg. fahr. As 3415 B.t.u. equal 1 K.W.-hr. of electric energy it will require about 2.5 K.W. heater capacity to heat 10 gal. of water 100 deg. in one hour, or 15 K.W. to heat it in 10 minutes. Thus the proper plan is to install as small a heater capacity as possible consistent with the required rate of heating water.

Sometimes it is desirable to connect an auxiliary electric water-heating drum into a hot-water heating system for use in the spring and fall, or as a booster in very cold weather. A few radiators can be connected to this drum and transferred to the main boiler when desired, or the drum may be in series with the boiler. When not used it offers no obstruction to the normal circulation.

We now come to a consideration of larger installations of electric boilers for hot-water or steam heating, or for power purposes. A large number of very successful electric boilers are in service in this country, Canada, and abroad. One power

plant has two 2500 h.p. boilers in service, and other units of 2000, 1800, 1600, and 1200, down to 30 h.p. are in successful operation at other points.

The design of an electric boiler is very simple. No tubes are used. The shell is a vertical steel cylinder with electrodes inserted through the upper head so as to extend downward into the water. Alternating current is used and passes through the water which forms a resistance and is heated by the energy dissipated thereby. The heat is not conducted to the water from hotter surfaces as in fuel boilers but is actually produced within the water itself. Consequently no temperature higher than that of the water can occur in any part of the boiler.

This is a very unique condition and means greatly increased life for boiler and an entire absence of scaling. Any sludge is in a soft condition and can easily be blown off. The boiler shell is not part of the electrical circuit, so that no electrolytic action affects it. Such action is confined entirely to the electrodes which are easily renewed. This is usually necessary only about once a year. The boiler and piping are neutral and grounded so there is no possible risk of electric shock even at the highest voltages. Boilers are in use in regular service on 22,000 volts, though any commercial voltage may be used.

A typical example of hot-water heating by an electric boiler is found in the Ford Motor Co.'s plant at Green Island, N. Y. This building is 1140 ft. long, 120 ft. wide, one story with a high central bay. The construction is of concrete and steel with over 60 per cent glass. It stands in an exposed position overlooking the Hudson River which supplies hydro-electric energy for power and heat.

An electric boiler of 3200 K.W. capacity at 4600 volts, 3-phase, 60 cycle, is used, and hot water is circulated by motor-driven centrifugal pumps at a rate of 1500 gal. per minute. The boiler occupies very little floor space and is located in the building itself among the other machinery, so that no boiler covering is needed, and there is no heat lost. The thermal efficiency is nearly 100 per cent.

To illustrate the small size and simplicity of an electric boiler in proportion to its rating it is only necessary to state that a unit of 320 B.h.p. is only 12 ft. high and 6 ft. in diameter and consists of a steel shell with from one to three electrodes inserted in the upper head and baffles to direct the water circulation. No stack, fuel storage, ash removal, or separate boiler room are required, and labor is reduced to almost nothing. No dust, fumes, or noise are made, and the efficiency is always nearly 100 per cent on all loads both for constant and intermittent service.

Electric boilers cannot compete in operating costs with fuel types except where very cheap hydro-electric power is available, or where off-peak power can be utilized with practically no extra cost. The investment and labor charge on an electric boiler are so small that it frequently pays to utilize surplus current in "off-peak" periods. Such a boiler may be floated automatically on the line to even up the load factor. For example, a mill may have unused power at night or whenever shut down which can be utilized for heating, or for steam production at a saving sufficient to pay in a few months for the installation of an auxiliary electric boiler equipment.

An electric steam boiler has a thermal efficiency of over 98 per cent. One K.W.-hr. of electric energy will make 3 lb. of steam. Assuming that 1 lb. of coal will make 6 lb. of steam every 1000 K.W.-hr. available current is equivalent to 500 lb. of coal per hour. Furthermore the labor of handling is saved. Thus it will be seen that under favorable conditions an auxiliary electric boiler is often very economical.

Whenever any water is flowing unused over a power dam an equivalent amount

of fuel energy is being lost, providing there is any possible use for steam or hot water within reach of the connecting power lines. Mills requiring lots of steam and hot water can often effect great economies by using all surplus water power to save coal and labor.

In view of the adaptability of electricity to every form of heat application and the constantly growing demand for a full use of the great potential water-power resources of this country, heating engineers must give increasing consideration to electric heating. It has only been possible in this paper to touch briefly the more important points and to give a few useful figures, but the subject is one which engineers will find worthy of careful study.

CONDENSED TABLE OF USEFUL ELECTRIC HEATING DATA

- 1-K.W.-hr. equals 3415 B.t.u.
- 1-K.W.-hr. will produce 3 lb. of steam
- 10-K.W.-hrs. equals one B.h.p. (standard boiler rating)
- 1-K.W.-hr. will heat 4 gal. of water 100 deg. fahr. in 1 hr.
(in thoroughly insulated tank).

DISCUSSION

A. M. FELDMAN: The statement that the "efficiency of electric heating is 100 per cent while that of the average coal heating is not over 50 per cent" is misleading because it is necessary first to burn coal to generate steam, then by mechanical means to transform it into electricity.

STEWART A. JELLETT: Most heating engineers have used electric heating to a very limited extent. The whole trouble so far with electric heating has been the excessive cost. If the cost of current can be brought down to a reasonable figure, then there is a big field for electric heating. In the Western countries, where water power is available, this is possible. In Seattle they are using electric heating. During the war electrical transmission was designed for a Canadian shell plant just outside of Toronto. In this instance 750 kilowatts were used in every blast furnace, and there were 16 furnaces.

My experience is that where hydro-electric plants are in service, if they must be supplemented with a steam plant, because of ice conditions, electricity is far from economical even compared with high priced coal.

LEE P. HYNES: The points made by Mr. Jellett are well taken. It is necessary to analyze all conditions thoroughly and get the right installation just the same as in any other type of heating.

PROBLEMS IN VENTILATION OF DEPARTMENT STORES

By A. M. FELDMAN,¹ NEW YORK, N. Y.

MEMBER

AN important problem to an engineer arises in the construction of a building which is to house a department store when he considers the system which is to govern the ventilation of the main floor and the basement. On these two floors, to a very great extent, are held the special sales which are so frequently conducted with the object of attracting as great a number of customers as possible. As a rule this means crowded aisles, a mass of people in a comparatively limited area.

The basement is generally windowless. Hence for its proper ventilation a supply of fresh air and exhaust system should be provided in evenly balanced quantities. On the ground floor one has an additional vital factor to consider, the inrush of cold air through the entrance doors during the cold season which causes the most unpleasant draughts. These are a most annoying causes of discomfort, especially to the sales force. On account of the draughts occasionally glass shields are placed in front of the show cases which are situated near the entrances. This does not, however, afford the clerks a sufficient protection. An investigation would easily prove that the sales people who work near the doors are all too frequently ill with colds or other respiratory diseases.

Entrances from without are usually provided with vestibules, and while these are heated, there is always a rush of cold air *into* the store resulting from the simultaneous opening of the outer and inner doors.

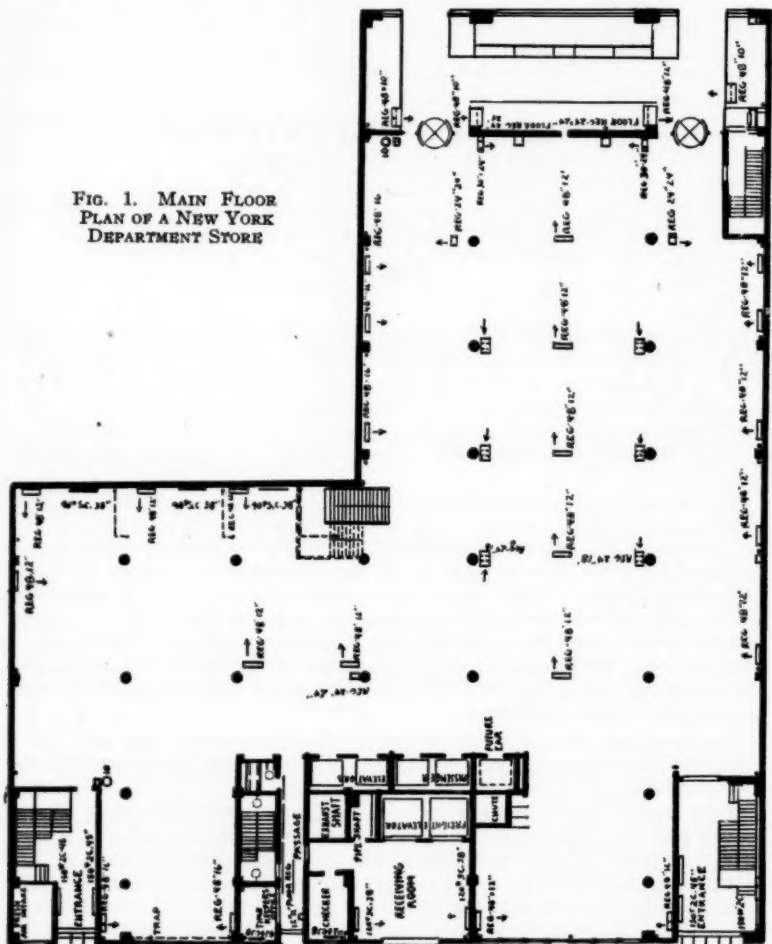
A practice developed by the writer of designing the ventilating system for the ground floor of a large department store so as not only to provide an ample supply of fresh air large enough for an unlimited number of customers and sales force, but also to eliminate all danger of the draughts above referred to. Highly desirable results have been satisfactorily accomplished even without the use of outer vestibule doors. The absence of these has added materially to the space surrounding the show windows, in consequence of which the prospective customers find added attraction in responding to the window-dresser's art.

This ventilating system is so designed that the fresh air supply is greater than the exhaust. More air is being delivered throughout the first floor than is taken

¹ Consulting Engineer.
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out by the exhaust fan. The excess of air has to escape through the elevator shafts and entrance doors. Therefore, when the entrance doors are opened, the warm air from the store escapes thus practically giving a warm greeting to the prospective

FIG. 1. MAIN FLOOR
PLAN OF A NEW YORK
DEPARTMENT STORE



customers. No draughts of cold air are rushing in on the sales force who are thereby in a better physical condition to give full attention to the customers.

In addition to this, the design includes the blowing in of re-heated air through registers in the ends of the counters at the doors, and also under the show windows just outside of the entrance doors. Notwithstanding the absence of outside doors

in the spaces between the show windows the atmosphere around the entrances is kept comfortably warm and pleasant, thus again inviting the street crowds to linger at the show windows, and adding to a still greater probability of their entering the store to buy the goods displayed.

This plan of ventilation has been successfully carried out by the writer in the Lindner Department Store in Cleveland, O., and lately in the L. M. Blumstein Department Store on 125th Street between 7th and 8th Aves., New York City, both types of the highest development of such buildings.

The problem of the distribution of air in a store is a difficult one on account of the large open floor area. The plan used by the writer in the Blumstein Department Store will be described.

Fresh air is taken from out-of-doors and warmed by passing over vent stacks and is discharged through a brick tunnel under the basement floor, then through the vertical galvanized iron risers, horizontal trunk lines and branches it is distributed under the furred basement ceiling. Space has been provided between the tempering and heating stacks for the future installation of an air washer which has been temporarily eliminated from the original plan.

Registers are evenly distributed over the entire ceiling and side walls for delivery of air to the basement, and branches are taken to the first floor through vertical ducts around the walls, behind the cases, discharging about 6 ft. 9 in. above floor and through registered panels in front of the show case bases throughout the center part of this floor.

The net cubical content of the basement is 253,000 cu. ft. and that of the first floor, 311,565 cu. ft. This is the space occupied by the fixtures and goods, as well as by the people taken into consideration. The fresh air fan is of a capacity to deliver a maximum of 100,000 cu. ft. of air per min., requiring a 50 h.p. electric motor to drive it. The supply of fresh air is equivalent to a complete change every 4 to 5 min.

During the winter months it was found that sufficient air is delivered without causing any draughts by running the fan 70 per cent of its maximum speed.

The out-door air is heated in winter about 74 deg. Fahr. and delivered through the registers at about 68 deg. Fahr., which was found satisfactory for maintaining the temperature in the store uniformly all day around 70 deg. The temperature is controlled automatically by means of thermostatic control. In the summer the fan is kept running at full speed.

The exhaust air duct work is distributed entirely through underground tunnels and branches under the basement floor and brought up through the floor in galvanized iron duct casings that are incorporated in the counters, the tops of which are utilized for the display of goods.

The exhaust ducts for the first floor are carried up along the building columns and are incorporated in the fire-proof protection of the columns. The registers are located in the cases near the floor.

A brick shaft built around the steel boiler stack is of such a size that the space is utilized for the exhaust air by connecting it to the tunnel. The exhaust fan is placed in the roof pent house, and the foul air is thus discharged above the roof of the building. The heat of the stack adds a great deal to the motive power of the exhausting effect over and above what the exhaust fan is designed to do.

Messrs. Robert D. Kohn and Charles Butler, New York, were the architects of the building.

DISCUSSION

STEWART A. JELLETT: Most department stores have open shafts passing up through the building and with the rush of air from the basement supply fans and the aspirating effect of the open light shaft, also with ventilators under control at the roof, so that they can be shut off when the building is not in general use, there is little difficulty in getting a sufficient amount of air out of the store. The aspirating effect from the heated store is very great. A revolving storm door is a poor thing for a department store although some stores still use them in addition to the swinging doors. The pressure of the incoming air supply plus the aspirating effect of the open light wells and shafts in most department stores will take off the air without running any exhaust fan.

A. M. FELDMAN: Mechanical exhaust is provided only for the center part of the first floor. The aspirating effect of the elevator shafts is depended upon for additional exhausting.

W. H. CARRIER: The use of fans, though it seems rather wasteful, is in practice warranted by the advantages. The difficulty with the shaft method of ventilation is that in warmer weather the induction effect is lowest, and as the weather gets cold the reverse effect results, so that the greatest induction effect often overbalances the supply at just the time when an excess supply is needed. It seems rather wasteful to use a fan for what might be termed a capacity meter, but that is about the effect that it has in controlling the exhaust. The fan necessarily has limited operation and induced circulation does not have any large effect when the fan is not running, which is relatively small. Therefore, the air quantity is somewhat in proportion to the speed of the fan and the amount of air that is being exhausted can be controlled at all times, and the exhaust and cleaning effect can be set just where it is wanted. A fan speed is easily understood and operated so that control is managed in a very desirable, mechanical way.

THE STATUS OF DOMESTIC OIL HEATING

By A. H. BALLARD, NEW YORK, N. Y.

NON-MEMBER

IN VIEW of the widespread interest regarding the use of fuel oil for heating houses and the many different minds that are at work on the equipment to be used for this purpose it is timely that this subject be thoroughly discussed so that we may know what the future holds in this field.

The greater majority of domestic burners are designed to use what is known as "Furnace Oil," a grade of oil having a gravity higher than 32 degrees Baumé and lower than that of kerosene. A burner of this description should be condemned because it is dependent upon a grade of oil that for the past ten years has shown a wide fluctuation in price, and, according to the opinion of reliable refiners, will show an equally great fluctuation in the future. Oil of this grade contains many elements which may be refined into higher grade products which are worth more money than this oil would bring if sold for fuel. The best type of burner is the one that can successfully burn the heavier grades of fuel oil both because the B.t.u. content per gallon will be higher and the cost of the heavier oil is considerably lower.

As to the supply of fuel oil it is axiomatic that so long as the world demands gasoline there must always be fuel oil. In the "Furnace Oil," of which we have spoken, there is enough gasoline to make it profitable to refine it for the gasoline it contains whenever the gasoline market rises to a sufficiently high point. On the other hand the price of fuel oil must remain relatively stable inasmuch as it has no higher potential value. There is no other field for its use that could command a higher price.

Therefore it is logical that the ideal oil for domestic use is one that runs from 20 deg. Baumé to 32 deg. Baumé gravity. There is a plentiful supply of this oil and a most important feature is the fact that it flows freely at zero and consequently does not require preheating, a requirement which precludes the use of a heavier oil than that we have mentioned.

Having determined the grade of oil best suited to domestic purposes the next consideration is the type of burner to be installed that will be productive of the best results. Because we have decided upon a light fuel oil, experience has shown that to achieve anything like perfect combustion this oil should be atomized, *i. e.*, disintegrated into minute atoms, instead of being vaporized, which, as the term would suggest, means the conversion of the oil through heating to an oil gas or vapor.

Presented at the Annual Meeting of AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, New York, N. Y., January 1924.

This latter treatment is not productive of the best results. The science of burning fuel oil successfully comprises the complete atomization of the oil plus the introduction of air in such a manner that each oil atom is completely surrounded with air that will insure its entire combustion. The equipment used to achieve this atomization should be of simple design and thoroughly dependable construction.

A most important factor to be considered in installing a domestic oil burning system is the fuel oil storage tank. This tank should be of a size that will contain at least three weeks fuel supply. The time may come when door to door deliveries of oil in five gallon lots will be made but these small scattered deliveries mean so much in the way of added costs that it would be most uneconomical to the buyer.

It must be remembered that the entire question of oil burning is one where service is paramount to everything else for, finally analyzed, the distributor is not only selling an oil burner or oil burning equipment, but he is selling a *fire* and to commercialize this, he must sell this *fire* to women and children. The distributor must always have trained forces on hand to render service before he may sell installations, so that in case of trouble, trained hands must be available to make the necessary adjustments or replacements.

With the domestic house-heating equipment, the automatic features are absolutely necessary to the equipment. A properly installed system must have maximum controls, whereby in a hot-water system the temperature of the water shall not rise above a predetermined temperature, and in a steam-heating system, the pressure in the boiler shall not rise above a certain limit; if these limits exceeded, the fire must go out. In connection with this, there should be room-temperature control, which is adjusted so that when the temperature of the room rises to a certain figure, the fire goes out.

Oftentimes, due to cheaply-installed heating systems where improper devices are used for releasing the air from the radiators, some part of the system becomes sluggish or slow, with the result that while the temperature might register 70 deg. in the living room, other radiators might be cold; the operator of the system observing this condition is naturally, led to decide that the fire should be burning when the automatic control has shut it off, and in most cases the system will then be tampered with to find out why the fire does not burn. The result is generally, that the system is put out of commission and service is necessary.

As previously mentioned the science of oil burning involves complete oil atomization and air introduction. It can be readily understood that there is no great mystery to oil burning, or at least nothing that is not well understood. The only qualification, therefore, that goes to make up an ideal domestic system is simplicity, factors of safety, and proper installation with proper service available.

There is a great deal said about the price of domestic equipments, and a great many people are making mistakes by putting in small storage tanks. This is practically the only item that can be cut down on that would not affect the working of the system, and, consequently, due to sales resistance the manufacturers are selling equipment making little or no recommendation for the proper size of storage.

To sum up recommendations for the consideration of an engineer for the benefit of those interested would be the following:

1. The system should burn oil of not lighter than 32 deg. Baumé fuel oil.
2. The burner should be of the atomizing type.
3. There should be no part of the equipment located inside of the boiler, so that in case of emergency, such as stoppage of the electricity, a temporary wood fire may be started to meet the emergency.

4. All dampers should be of a size so they cannot close more than 80 per cent of the area of the smoke pipe.

5. All equipment installed with an oil-burning system should be of the very best materials and workmanship, securely fastened, all parts in a convenient place, and so constructed that it can be kept clean and frequent supervision made thereof.

6. The question of noise, both mechanical and combustion, should be taken into consideration in any type of an oil burner, because the system may be very quiet in one place, and in another the effect may be offset, and the vibration may be heard throughout the house.

The matter of noise is a problem that all manufacturers have confronting them every day and they can overcome it when they reduce it to practice in each individual instance and put an expert on the job to analyze conditions and adjust the system.

In conclusion, when a comparison of value is asked for between oil as a fuel and coal, virtually it requires 156 gal. of oil to equal 1 ton of coal, when a given percentage of efficiency is used in both cases and where a fair standard of B.t.u.'s in both coal and oil are taken. However, in ninety-nine cases out of a hundred, the comparison shows much more in favor of oil due to the fact that the furnaces for coal burning are so varied that it is astonishing how crudely coal is burned in the majority of cases.

Statements have been made by manufacturers that 100 gal. of oil equal 1 ton of coal. This is not true in the case of heat values. However it is a fair assumption to say that 135 gal. of oil will equal 1 ton of coal when reduced to practice in each domestic unit properly installed.

The questions of virtue of oil in the way of saving labor, ash removal, and dirt as shown by salesmen, are points well taken. The consumer properly equipped with an oil-burning system would suffer in the way of heating costs a great deal before he would change back from oil to coal.

Oil burning as applied to residences has not even started as yet and the future will show a rapid growth for the manufacturer who reduces to practice a quiet, effective atomizing system and renders service thereafter.

DISCUSSION

J. M. SEWELL: I would like to ask Mr. Ballard if he has had any experience in heating a residence and a garage in conjunction with the residence where they formerly had coal and either hot water or steam, the garage being heated to around 40 deg. and then putting in an oil burner of the type that positively shuts off when the predetermined temperature has been raised in the house.

A. H. BALLARD: Wherever there is an automatic system installed in a residence and garage depending entirely upon automatic control, it is perfectly obvious that if you are controlling a temperature for the house, it is necessary to have an individual unit for the garage. The same storage tank can be used but there must be two separate heating units. The garage can be very much colder than the residence.

I have equipped large estates having a greenhouse and a residence and ser-

vants' quarters and different problems of that kind. I put in one storage tank and put in separate units which are very expensive and erect each one individually. If I want this room 60 deg. and want the house 72, they both will remain right there and they will never change. I can take a little piece of ice and go right up to the regulating thermometer and just as soon as I touch the ice to that the fire goes out and if I take the ice off again it will start up. Your curve line is as straight as you can shoot a rifle.

The following case is typical. A man in Boston had installed an oil-burning system in his residence and we told him for economy to set his regulator at 70 deg. during the day and 50 deg. at night. He followed this plan for a week and then left it at 70. Someone that we had referred to him asked him how much fuel he used and he said, "I don't know, and I don't care; it's the only time I have lived. My house is just as if I were living in Florida. I ask no questions of anyone. I go into my bedroom, open my windows, go to bed, get up in the morning, the temperature is 10 or 12 deg. in the bedroom. I walk into the bathroom and it is 72 deg." Those things are all predetermined. You can have any temperature providing you are not interlocked.

M. W. EHRLICH: I believe the author has talked only on the power driven units, as of course most of us know that gravity fuel oil burning is out of the question as an engineering proposition. Mr. Ballard mentioned some quite high efficiencies with oil and I raise the question whether those efficiencies are obtainable by merely installing oil burning equipment in existing domestic coal burning heaters.

It was stated that in at least 50 per cent of the cases, with which Mr. Ballard has come in contact, the steam circulation is poor and I would agree with him on that score, but I take issue with him, however, on the point that he is under the impression that those are among the jobs designed by heating engineers. They were jobs merely put in by steam fitters.

Mr. BALLARD: I said I was glad to talk to a gathering of this sort for the reason that the majority of the heating systems that are put in are not designed by heating engineers. I mean you will go out and find steam fitters putting in jobs without the least recommendation and their only experience is as a steam fitter.

When I say fifty per cent of those, what I meant by that was that they are not fifty per cent all crudely wrong, but that there are some corrections to be made.

Years ago heating engineering was my business, and I am in sympathy with you for I know what you are trying to get at and that your intentions are of the very best, but I say this: You don't control the entire industry and if more people listened to you the whole scheme of things would be a whole lot better off. We find undersized boilers in a great many cases which are not planned by heating engineers, and undersized pipe sizes are not installed by good heating engineers. I can show you a number of radiators in living rooms today where the cold air to supplant the hot air has got to travel clear across the room. I am not criticizing this body only I want more cooperation so that when we get to the point that you are up to strictly oil heating, you will have beautifully working systems.

M. W. EHRLICH: I am glad we understand each other on that point. There is just one other factor that I would like to bring out from my own experience as well as many others with whom I have talked on general heating practice. I believe a good many of us know that temperature regulation alone, particularly by a ther-

mostat in one room, does not answer the problem of regulation, because there must be pressure control with it to get a certain regulation. If that is true in a coal fire, it would be more true in an oil fire, which is intense while it is going.

H. M. HART: Couldn't we save time by following our usual custom—of getting all these questions at once rather than spend so much time on the discussion of each question? I believe that we are devoting too much time to this one paper. While it is interesting, I think there are a lot of questions we would like to ask, yet on account of the time that is being consumed I am afraid I am not going to get a chance.

H. W. BROOKS: The Bureau of Mines is possibly in position to confirm Mr. Ballard's statement that the era of domestic oil burning is here. We are advised by the information section of the Bureau that there are more inquiries being received on domestic oil burning at the present time than any other single branch of fuel burning practice. We are not unmindful of the public service which might be rendered in an investigation of domestic oil burners.

It is a public service for which there is an insistent demand and the Bureau is intensely interested in the subject.

Oil burning today lends itself to close regulation in a way commensurable with no other form of fuel burning practice with the exception of gas. The possible close control of CO₂ and the close control of excess air make this system almost unique in fuel burning practice at least as compared with coal.

J. M. SEWELL: In regard to my question about a separate unit for a garage, heated in conjunction with a residence needing 40 to 60 ft. of radiation, what I want to learn particularly from Mr. Ballard is whether he would recommend a separate oil burner or a very small boiler, oil burning, costing around \$350 or a separate coal heater for that particular garage.

Mr. WESCHLER: It has been reported that oil burning causes heavy, cast iron sectional boilers to crack. I should like to know if Mr. Ballard had any experience along this line.

HOMER ADDAMS: Are these oil burners easily adaptable to all forms of furnaces or boilers as at present designed and constructed?

H. C. EICHER: Pennsylvania has 14 normal schools which carry large heating plants. There is a continuous load on these plants and the Department of Public Instruction has been making an investigation for several years as to the best way to maintain that continuous load. Can Mr. Ballard give some assistance, some suggestions or recommendations which by installing oil burners the State of Pennsylvania can realize any considerable saving, and in those 14 large plants would it be justifiable to replace the present coal installations?

E. B. LANGENBERG: Several questions occurred to me. Is the automatic valve for shutting off the oil reliable? Is the ignition system dependable? Is there any flexibility in the burner? Can it be run low or high? Can the correct (analysis) oil be secured at all time for any particular burner? What effect is it having on either cast or steel furnaces?

The statement has been made that the intense heat—2000 deg.—either melts the cement at joints or powders it and blows it out. What effect is the sudden high temperature having? Do the needle valves clog with carbon? Are they easily

accessible for cleaning? Will oil burners be successful in apparatus now in use and as now designed?

I would suggest this organization pass a resolution and present it to Mr. Ballard to be presented to the *American Oil Burning Association* that they form their own organization as an industry itself and establish their own research bureau. We have come to the conclusion in our own advisory committee that apparatus to use oil, gas and other fuels, must be designed for that particular fuel. They can be adapted, to the present apparatus, but it is a difficult proposition.

H. M. HART: Mr. Ballard spoke of the proper mixture of air and oil. Is there any way a layman can determine when he is getting the proper mixture of air and oil?

Another thing that is important, I think, that he offered in the way of advice to heating engineers or contractors, recommending the installation of oil burners, is the fact that an oil burner is noisy in its operation and a great many of them are removed because they have been installed in boiler rooms that are not suitable for an oil installation on account of the location and the means for conveying the noise through the house. I think it is quite important in designing a house now to think of the location of the boiler room so as to locate it in as isolated a place as possible and have it insulated for sound so that there will not be that objection in the installation of oil burners.

H. B. HEDGES: I would like to ask whether there has been any satisfactory type developed for automatic operation in the isolated communities where they don't have gas to operate the pilot valve, which, I believe, is usually required.

F. C. BARTLEY: I might ask Mr. Ballard what efficiencies can be gained by utilizing the heat from the flue products for preheating the air for atomization of the oil.

J. F. MCINTIRE: I would like to ask Mr. Ballard if in the application of an oil burner to a coal-fired boiler he would recommend a smaller size of boiler than would be selected for the same job where solid fuel was going to be used for burning.

IRA WOOLSTON: I would like to hear something about the fire hazard in connection with present domestic systems.

A. H. BALLARD: Regarding the fire hazard, that is taken care of in this way: If a system is approved by the Board of Underwriters and approved by the district chief having jurisdiction, that is, it complies with the regulations, the insurance rates do not increase and the fire hazard is practically nil.

Regarding the size of boilers, naturally that is the problem presented with coal, but it also applies to oil. Greater efficiencies can be obtained at 125 to 150 per cent of rating for boiler, if that answers the question. In other words, you get less radiation loss overfiring the boiler than you do underfiring.

Preheated air is very advantageous provided it isn't overdone. Naturally your capacity for air has got to be larger when heated and your cubical content is less when heated than cold. We always heat air where it is practical.

We have two ways of doing away with noisy fires. That is the greatest problem we have today. In any oil system that is installed in the basement of a house where we have our pipes and our pumping we use no gravity flow whatsoever. I condemn that method. In pumping the oil you get a vibration which is telephonic

providing you touch another piece of pipe or anything that will carry sound. It has been my pride that in residences without a plastered ceiling I can put in a system and the noise will not be objectionable. In each case it has to be individually engineered for that purpose to keep away the noise. Combustion noise is not objectionable. If you get excess air you get a very fast fire which gives you a roar, and it is noisy and creates a vibration from your boiler which telephones right into your radiators. That is in vaporization systems and not in atomization systems. Noise, however, can absolutely be avoided.

Proper mixture of air and oil is a scientific question and no layman can tell that unless you analyze stack gases. If you put on a CO₂ outfit, set the dampers for general conditions. We have a maximum fire with no air admission except what is admitted through our system. If your atmospheric conditions vary your efficiency increases or decreases according to your atmospheric conditions.

Regarding the effect of oil on boilers. Localization of heat is the only thing that will cause trouble inside of a boiler. In a great many cases an inexperienced hand will introduce his air in the wrong place or take in air. Oil will not accept air after a certain temperature and that air will travel and form a regular pocket around your fire. I have seen inefficient fires where the air will come around and cool a section and crack it. I want to say this, however, that a properly installed oil fire is not as injurious to any type of a boiler, for you have no stoking.

Bricking of boilers is a problem. Different bricks have different fusing points. Different clays have different expansions. In my work all clay has been eliminated and we use nothing but a bath, something that has the effect of glueing the surface. We use high temperature cement only for repairs. In a low pressure job I have bricks that have been in as long as eight years. In others I have had to rebrick in six months. That is a question we can't tell anything about, but we do the job and don't say anything about it.

The reliability of the control depends upon a piece of mechanism made by man which is not infallible. The only thing we do in case the control fails and liquid oil should go into the boiler, it immediately shuts off the oil supply so that the hazard is taken care of.

The advisability of replacing coal with oil depends entirely upon three factors. First, the amount of coal you are burning; Second, the cost of coal; Third, your requirements. In a great many cases we find that they would have to increase the size of their boiler plant if we don't put in oil. By burning oil we take a percentage of rating greater than they can take off with coal and it saves them buying new boilers. In recommending oil burning we first analyze the man's particular condition, advising him how much money he can save.

I have been asked about types of boilers but I can't mention them except to say that for the State of California I made a test of all the cast iron sectional boilers made and lined them all up and evaporated into atmosphere. The results varied 20 per cent.

Hot-air furnaces are things I have condemned after using many in California some years ago.

On the question of garages, if the garage is connected up to the same furnace as the residence, it is just as well to leave it on the system, because when the house requires heat so also will the garage and then it is a question of manual control and the amount of radiation is so small there that it won't be worth considering.

One thing about oil, we don't like jobs that are too small. You can't apply a hot fire to any unprotected parts, without it being fire brick, and if you have a very small furnace and you have to brick it up, you have no space left for the combustion chamber, consequently you can't burn oil very well.

Regarding the efficiencies that you are asking about here on domestic installations, I made about ten tests and The Standard Oil Co. made the same tests. Their man got 14.6 lb. of water on ten straight installations.

Regarding the public service, it is absolutely ridiculous to sell a person an oil fire where it is sold in a residence and the man goes off to work and has children running around unless you can give the customer service. You have four controls on one of these jobs and a maid or a woman can go through with a duster and knock back the temperature and the fire won't come up. They think there is something the matter with it and throw the whole thing out of kilter. If you don't give them public service you have given them nothing. I don't accept an installation outside of twelve-mile radius in a city where we have a branch, unless it is some very large case or something very unusual.

THE SEMI-ANNUAL MEETING, 1924

GATHERING from all parts of the country, members of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS convened at Kansas City, the "Heart of America," for the 1923 semi-annual Meeting, held at the Hotel Muehlebach, June 10 to 12.

Five business and professional sessions were held during the three days, the salient feature of which was the presentation and discussion of the Code of Minimum Requirements for the Heating and Ventilation of Buildings. The Meeting convened at 10 o'clock, Tuesday morning, June 10, with Pres. Homer Addams, New York City, presiding. Walter E. Gillham, President of the Kansas City Chapter greeted the Society and John S. Cannon, City Councilor of Kansas City, representing the mayor, officially welcomed the visiting members of the Society to Kansas City.

After a short resume of the progress of the Society during the past six months, President Addams called for reports of Committees. The session was then given to reading and discussion of papers.

At the suggestion of E. S. Hallett, St. Louis, the following resolution offered by W. H. Driscoll, chairman of the Committee on Research was adopted by an unanimous standing vote:

WHEREAS: The investigations being conducted at the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in connection with the problems of the comfort and health of human beings have been productive of such favorable results, and

WHEREAS: The results of these tests represent a distinct contribution to science and will prove of the utmost value to the present and future generations, and

WHEREAS: These investigations were only made possible by reasons of the generous and sympathetic cooperation and assistance of the executives of the U. S. Public Health Service and the U. S. Bureau of Mines, and through the energetic efforts and self-sacrificing spirit of the scientists engaged in these investigations be it therefore

Resolved: That this Society, in session at its Semi-Annual Meeting in Kansas City, Mo., show its deep appreciation of the great work accomplished by extending a rising vote of thanks to the following:

Dr. R. R. Sayers
Mr. O. P. Hood
Dr. W. J. McConnell
Mr. F. C. Houghten
Mr. O. W. Armspach
Mr. C. P. Yagloglou

and be it further

Resolved: That a copy of this resolution be spread on the minutes of this meeting, and that copies be furnished to the press.

The Research Session was conducted by W. H. Driscoll, who gave the following report of the work of the Research Committee:

Address of W. H. Driscoll

I haven't come here with a glittering array of facts and figures to confound you, but I am here to talk to you on a subject that is very close to my heart. This is a family affair. We have our family problems, and at times when the rent is due, or the interest is due on the mortgage, or the income is due, I have to sit down with my wife to discuss the condition of our finances, and to consider ways and means of meeting the situation that confronts us, and we have to find a way of somehow living through the period of depreciation that is before us. So we are going to assume the same position here today. I am the husband, and you are the wife. We are going to sit down and talk about it. We are going to find out how we are going to get through. We are not going to be divorced or sell the old homestead. We are going to live through the situation as we have lived through such conditions in the past.

I was very much encouraged in reading an article in a Chicago newspaper the other day, which said that it was a matter of record that all the great movements such as fraternal organizations, charitable organizations, and movements of that sort were during the first five years of their lives on the uproad for a part of the time, but at the end of the fifth year they reached the very depths and then came back here. That gave me a great deal of courage when I realized that we were at the end of a five-year period in connection with the Research work, and that the conditions that confronted us were not at all unusual, and that there were reasons why our bank account should be somewhat depleted, and that there were reasons why we were up against a condition now that might be very discouraging if we permitted ourselves to become at all despondent in connection with it. Twenty years ago I joined the Society. My election to membership was announced at the Society's Summer Meeting in Detroit in 1904. At that Semi-Annual Meeting only 11 members registered. The Society had been in existence 10 years. It had come into being through the courage, vision, and foresight of men who were not schooled in the technical knowledge of our business. Many of them were men whose education was somewhat limited, but they had a vision of the need of such a Society as this, and despite their limitations they gathered together men of the same vision from all different parts of the country. Year after year they struggled in order that the vision that they saw might be fulfilled, and in order that the organization that they imagined might come out of that start would come into being and fulfill all the promises and all the things that were expected of it. After 10 years 11 members responded and registered at the Semi-Annual Meeting. You know those might have been dark days in the history of this Society. I suppose men with less courage than the men who still remained in it might have thrown up their hands and said, "What is the use? Nobody wants this. We are putting our time and our energy into it. Why should we concern ourselves with an organization in which the people of this country, the engineers in particular, have no interest?" Nevertheless they stuck to it. They went down into their pockets, and they paid the expenses far and above the ordinary dues that they had established, and they carried it through those critical days until today. While we may not numerically be equal to some of the great engineering societies in the country, I say that, as an engineering organization, as an organization devoted to the work of its own kind, as an organization that has fulfilled in every way the greatest hopes of its founders, we stand second to none in this country, and I express that opinion with a full knowledge of the great work that is being done by the greater organizations. I know the feeling that they have for us, and I know they recognize the importance and value of this heating and ventilating Society, and the position that it occupies in the engineering world.

World Recognition of Research Work

One of the great things that the Society has accomplished that has brought it recognition, not only in this country but all over the world, has been the Research Laboratory. And I say that the men who are responsible for bringing that Research Laboratory into being, who are responsible for having carried it on to the point where it is, were men of courage and vision equal at least to the courage and vision of the founders of the Society. They put it on a basis to carry it through a period of five years, and the end of the five years is here, and the enthusiasm that they injected into it at the start has waned to some extent. The financial returns are not as great today as we might hope for various

reasons, some of which I will try to outline, but the Laboratory has become a recognized institution. The Laboratory has become the bone and sinew of the Society, and to think that at this time those of us who have received the benefits of membership in the Society, those of us who have seen the Laboratory proceed so successfully in the things it was supposed to do are not going to fall down on our job, because I am sure we are just as courageous as old Andrew Harvey, William J. Baldwin, and Rolla C. Carpenter, all of whose names stand out boldly in the history of this Society. We are not going to let the Laboratory die. We are not going to give up our homestead, as I said before, and sell our home and go out of business and divorce ourselves from this great institution. We can accomplish too much to quit at this particular stage.

Personally, my knowledge of what the Laboratory was doing was somewhat limited until about the first of February. I was a member who had confidence in those members who were promoting the Laboratory, and in those who were responsible for bringing it into being. I had such confidence in them that when they came to me and said, "We want \$5, \$10, or \$15" I gave it to them, because I thought it was the proper thing to do.

In January of this year some members of the Society asked me to take over the Chairmanship of the Committee, and I said, "No, I would not do it." I had the most common reason that any man could have, the reason that we all have, the reason that we always give for not doing the things that we ought to do, "I am too busy, and I can't give it the time that it requires, and I am not going to take it over." They came again and said, "We will help you. We will jump in and help you, and there isn't anything but what we will do to help you out. You needn't do any work." I said on top of that fact that I was too busy; that I didn't know what the Laboratory was doing; that I couldn't go out and sell it to anybody; that I didn't have the vision and imagination for that sort of thing; that I would not be satisfied to take hold of an institution that did not have the proper spirit of enthusiasm and try to whip it into shape; and that I was not going to spend the time on it. They said, "Well, we have looked the field over. We have decided that the men who have carried this thing through up to the present time are not to be asked to carry it on any longer, and it is the unanimous opinion of the Committee that you should be the Chairman." They told me if I refused they would be in a rather critical condition; that there was only one thing left to do, which was to go into a meeting and make a resolution either to continue or abandon the Laboratory. I said, "I will tell you what I will do. I have not dodged any work that has come to me in the Society, that I felt I could accomplish reasonably, and if you go in and make such a resolution, and the reaction in that meeting is such as to indicate to me that the Society wants the Laboratory, I will take over the job, no matter how big the sacrifice may be."

Society Wants Laboratory

Many of you were there when the motion was made, and the reaction was all that any one could expect, and I was forced to redeem my pledge and take the job over. Then, I said, "Well, we only have the 'Old Guard' here anyway, the same limited number who were interested in bringing it into being, and the fellows in the byways and high-ways are not interested, and unless we can get a greater interest throughout the Society, we might just as well lie down on the job, and so I sent a letter on February 6th to all the members of the Society. Within 24 hours there were 50 replies on my desk, and for the next two months the replies came in from every town in the country, from every state in the union, from the provinces of Canada, from foreign countries all over the world as far away as Shanghai.

Six or eight hundred letters came in and swamped me and my stenographic force. Among those six or eight hundred letters not more than five showed the slightest opposition to the Laboratory. On the contrary the enthusiasm for it was tremendous. Member after member said, "If the Laboratory shuts down we are through with the Society."

"We had better give up the Society than give up the Laboratory." The most enthusiastic letters came from those fellows situated away from the great centers—the fellows out here in Kansas, Colorado, and down in Oklahoma, the fellows up in Vancouver, over in Shanghai, and from the stricken countries of Europe were the most enthusiastic for the Laboratory. These men in the far distant lands knew every movement that had been made in the Laboratory, and they were not only willing that the Laboratory should continue but they were offended to think that anybody should think otherwise.

A letter from Dr. Alice Bryant of Boston, one of our leading members, was the grandest of all, and I say that without reflecting in any way on other splendid letters that came in. It was so complete and so comprehensive and so conclusive, that I saw

fit to have it immediately published in the JOURNAL in the hope that the members might read it.

You know I have never pretended to be much of an engineer. My training has been in business, so my first thought was to find out where we stood financially, and I believe that we ought to call a "spade a spade," lay our cards on the table, and have an understanding of the problem.

The five-year pledges that many of the manufacturers made have expired, and they have not done the great work that the members of the original Committee on Research performed. The pledges expired and the manufacturers have not responded. They thought when the five-year pledge was over the Society would have established some means of maintaining the Laboratory other than asking for a continuation of that five-year pledge, and because many of them were not doing as large a volume of business as they had expected, they said, "No more money." The group of manufacturers who had contributed \$5000 said, "Nothing doing. We are through." And the sources from which we had obtained the funds to carry on the great work which we had done were dry. The manufacturers said, "What does the Society do to maintain this?" The answer is, that some members of the Society do a great deal, but the Society, as a whole, has not and does not and never has responded as well to the demands, the financial demands, of that Laboratory as it should.

Practical Value of Research Reports

The Society as a whole, perhaps many of them, had the same idea as I had, that this work is more or less theoretical and has no practical value, consequently it is not an easy thing to sell. Now, right there, I discovered something that I had not appreciated before. "What does this research material amount to? Why don't they put it in some form so that we can use it?" Simply because that is not the function of the Laboratory. The Laboratory investigators are not men who are up against the practical problems that you and I are up against. They are down there to dig out the facts that we need, and it is up to you and me with practical minds to fashion those to our own needs, and to make them of practical, usable, workable value.

So a Committee has been appointed. Alfred Kellogg of Boston is Chairman of that Sub-Committee and he has gathered a Committee of high grade men from the Massachusetts Chapter, and they are now making a study of all this great array of literature that the Laboratory has turned out—50 or 60 splendid papers—many of them deal with the same subject of course, and that has been one of the criticisms. There is bound to be repetition. They may be working on a problem for years, and every year they have to bring out a report about it and tell us what they are doing. "Same old bunk!" say the practical fellows. I am giving you the reaction, not of the thinking men of the Society, not of the men who really have an appreciation for these finer things, but of the men who have to be reckoned with just the same. Mr. Kellogg's Committee is going to make a study of that critical feeling.

Now, we have had a very, very small sum of money in the Treasury—a very small sum. When I took the job we had \$3000 or \$4000 in the Treasury, and we had a budget of \$30,000. We ought to spend \$50,000, and we might spend \$100,000. We could spend far more than that if we had the money, because, after all, what we accomplish depends on the amount of money that we have to spend. We have been running along for about five months on a budget of approximately \$30,000 to \$32,000. We had a deficit facing us unless we curtailed the program, or unless we immediately received the funds that we needed. We were unfortunate in starting over on this financial program. This thing should have been discovered a year ago, but somebody like myself was too busy to give the proper attention to it.

The reports lately have been far more encouraging, and the manufacturers are beginning to see the light of day, but, after all, it is a very repulsive thing for me to go out with my hat in my hand and say to the fellows who make traps, boilers, etc., "We need some money." No matter how worthy the purpose may be, I was never any good in getting money for the Red Cross and that sort of thing. However somebody has to do it. Because it is repulsive to me I don't want to be Chairman of a Committee whose members have to go out and do that, but we have to meet conditions as we find them, and that is what we have to do at the present moment. In the meantime we must establish this Laboratory on a basis that gives it a fixed and definite income—a basis on which we can establish a basic program of procedure. We must have an income that will come in regularly every year regardless of business conditions, and regardless of the attitude

of the manufacturers generally, so that funds must come from the members of this Society.

In my last letter to the members I called attention to the fact that there is going to be a resolution introduced at this Meeting asking for an increase of \$10.00 per year in the dues of members which is to be deposited in the Research fund and devoted exclusively to Research purposes. We have 1600 to 1700 members. It gives us a basic fund of \$16,000 or \$17,000 per year. There may be some opposition to that, but I don't think so. I don't think there is going to be any real opposition to it. It may be necessary, unfortunately, for us to lose a few members. I am convinced that you members here before me today, and the hundreds who are elsewhere are back of me, and I know that the response to that suggestion is going to be overwhelmingly filled. This plan gives us a fund on which to start. Where are we going to get the rest? Are we going to have drives every year? Not if I can help it. We must get the money in other ways, and one of the most feasible is THE GUIDE. It is established and is growing, and we should get \$10,000 from it without the slightest trouble. A substantial amount is expected from THE GUIDE fund this year to help out.

We need money badly right now. We can hardly go through the year with what we have including what was left over from last year, and I would consider it a great calamity if we were not in better shape at the end of this year than when we started the year. I would consider my term as Chairman to be very, very unsuccessful, if we could not make both ends meet, and we had to draw on our capital account. It would be heading towards bankruptcy, that is all. But if we carry through this year and establish a policy that will fix forever the method of financing our Laboratory, why, we shall at least have met some success.

Most of you are familiar with or have read the resolution passed at the January Meeting in connection with the matter of testing which has come to be called commercial testing. We have had some objection to the question of testing, but in connection with it there is a chance to help finance the Laboratory. The testing that we do there under the terms of the resolution would carry with it a sufficient income to extend our fundamental research work which is the specific purpose of the Research Laboratory. The testing that we have considered a necessity is simply a means of acquiring additional funds for the purpose for which the Laboratory was established.

Now, many of you know the uproar in the heating industry, by disparity, of the boiler ratings. Professor Dibble looked into this question of testing boilers to take the heating industry out of its present chaotic conditions. Invitations were sent to the cast-iron boiler manufacturers to meet in Pittsburgh and talk this situation over. Twenty-three cast-iron boiler manufacturers were represented, and thirty-six representatives of our own Society, and they proceeded with the idea of our going into this question of testing the cast-iron boilers. That is one of the activities of the Research Committee at the present time. I am not prepared to make a report, but out of that initial start, we hope to accomplish a great deal of good for the heating industry—a great deal of good to stabilize the conditions that are connected with boiler rating.

I wish to say that this session will be devoted to Research papers prepared at the Laboratory as a result of the investigations made there. The papers to be read here will be presented in abstract form so as not to take up your time, and we will try to avoid getting into one of those conditions of disinterest that sometimes comes by reason of the character of these papers. They are not always the most interesting.

W. S. Timmis, chairman of the Committee on Research Endowment Fund, submitted his report which was read by Secretary Houghten as follows:

Report of Committee on Research Endowment Fund

I beg to acknowledge your request to serve on the Endowment Committee of the Society which, as stated by you, has for its objective, the raising of an Endowment Fund, the income from which will be sufficient to defray the expenses of the Bureau of Research, and I beg to state that I shall be pleased to serve on this Committee, but I wish to give you my reaction to the general idea of an endowment in connection with the Bureau of Research at this time. I do not think it will be practicable during this period of business depression to go out and secure funds in an amount sufficient for the purpose as discussed at the Annual Meeting in January of this year. I have discussed this question with a number of members of the Society and their reaction has been coincident

with my own, and I have been invariably informed that it would be imprudent at this time to make an endowment drive.

The thought has occurred to me however, that this Committee might consider a suggestion which is not entirely new, but which was discussed in the early days of the formation of the Bureau of Research. At that time thoughts were expressed that there possibly would be organized by the Government a National Bureau of Research of which the Bureau of Research of our Society might be a part. I will therefore suggest that this Committee, or a Committee to be appointed, be authorized by the Society to approach the Secretary of the Department of Commerce and the Secretary of the Department of the Interior with a view to forming a National Bureau of Research in connection with the existing Bureau of Mines, and Bureau of Standards. This National Bureau of Research should ultimately cooperate with the founder engineering societies such as *American Society of Mechanical Engineers, American Institute of Electrical Engineers, American Society of Civil Engineers, American Institute of Mining and Metallurgical Engineers, Chemical Engineers, Refrigerating Engineers*, and other Engineering Societies, and each of these Societies should direct and plan the work to be conducted at the National Bureau of Research, and should contribute some proportional share of the expense of conducting such research as will fall within the scope of their activities. Such an arrangement should meet with ready response from Secretary Hoover who is the first engineer who has had a prominent cabinet office for many years, and who has exhibited a profound interest in the part engineers are playing in the development of the country and its research.

If such a proposal were carefully prepared and submitted to the Secretary we might even go so far as to state that we will contribute, say, \$20,000, per year, it being understood, however, that we are to lay out the program and direct the work which is to be done.

We have a splendid record of achievement upon which to base this request, and our Society has blazed the way for a National Bureau of Research. If such a movement can originate with this Society, we should undoubtedly have great cause for gratification, for we will have rendered a great national service and perhaps even an international service, in that we know that whatever this country undertakes to do along these lines will be done properly, thoroughly and efficiently.

This, Mr. President, in brief is the suggestion I would make in response to your honored request that I serve on the Committee on Endowment.

I regret exceedingly that it will be impossible for me to be with you in Kansas City at this Meeting, but I can assure you that I am with the Society, and consequently with the Bureau of Research both heart and soul and you may count upon me to do whatever lies in my power to help the Bureau of Research and the Society.

I feel that the suggestion above can be worked into a reality to the good of the country and to the glory of the Society.

Respectfully submitted,

W. S. TIMMIS, *Chairman*

W. H. Driscoll then submitted the following resolution:

Resolved: That the Secretary be instructed to draw up a proposed amendment to the Constitution, providing for an increase in the dues of members and associate members, of \$10.00, said increase in dues to be used for research only, and that he be guided by the present Constitution and By-Laws in presenting the amendment to the members, the amendment to be voted on at the next Annual Meeting in January, and to become effective as soon as possible.

Presentation and discussion of the Code of Minimum Requirements for the Heating and Ventilation of Buildings, led by L. A. Harding, Chairman of the Code Committee, took part of the time of the second, fourth, and fifth sessions.

E. B. Langenberg, St. Louis made the following motion which was seconded and carried:

I move that the Standard Code as now submitted by the *National Warm Air Heating and Ventilating Association* be approved by this Society.

The formation of a local chapter at Seattle was announced at the last session of the Meeting.

The social program for the Meeting was crowded with events. Whenever there was a spare moment from a professional session, the Entertainment Committee were ready to start the guests on a sight-seeing trip or escort them to a luncheon, musicale, golf meet, banquet, or dance.

Following the session on Tuesday afternoon, 35 cars with a motorcycle police escort took the visitors for a 30-mile drive, stopping at Mission Hills Golf Club for dinner.

A special luncheon was served to members of the Society on Wednesday at the Kansas City Athletic Club, followed by a golf meet at the Meadow Lake Country Club. A dinner followed the meet. Those not playing golf were taken on a trip of inspection of the Stock Yards, the plant of the American Radiator Co., and the power plant of the Kansas City Heat, Light, and Power Co.

A banquet and dance held at the Hotel Muehlebach on Wednesday evening was the big social event of the Meeting.

The women guests were provided with special entertainment. An informal luncheon was served for them on Monday and on Tuesday an automobile ride was followed by a special luncheon at the Hillcrest Country Club. On Thursday they attended a luncheon and musicale at the Roof Garden of the Kansas City Club.

In token of the appreciation of the members for the hospitality and entertainment furnished by the Kansas City Chapter, the following resolution, offered by E. B. Langenberg, was adopted:

This is the first opportunity that the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS has ever had of being the guests of the Kansas City Chapter. It is also the first time the meeting has ever gone as far west.

After some hesitation the Council of the Society finally decided to accept the invitation of the Kansas City Chapter. It was naturally expected that a smaller number of people would be in attendance on account of the distance from the large eastern cities, but the number present has far exceeded all expectations.

We were aware of the tremendous enterprise of this great western city but we were not prepared for the wonderful reception which has been so graciously extended to us.

The AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS assembled in its last session desires to express to the Kansas City Chapter and also to the committee of ladies cooperating with the Chapter, its sincere appreciation for the many courtesies which have been extended to the members and especially to the visiting ladies.

It is with a feeling of deep regret that we are finally compelled to take our departure. The slogan "The Heart of America" had only a vague meaning to those of us who had previously heard it spoken. We know now that this forceful and imaginative expression has a real true meaning.

In the years to come every visiting member when his mind reverts to this occasion will visualize a city of rare beauty, tremendous force, and a place of delightful homes teeming with people whose hearts are responsive, and who have a delightful consideration for those who are fortunate enough to tarry here, if only for a few days.

Be it therefore resolved:

That these expressions of our appreciation be spread upon the minutes of our Society and also that copies of these resolutions be forwarded to the Kansas City Chapter and also to the local press.

KANSAS CITY, June 12, 1924.

PROGRAM SEMI-ANNUAL MEETING 1924

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

HOTEL MUEHLEBACH, JUNE 10-11-12, KANSAS CITY, MO.

June 10, 1924

MORNING—BUSINESS SESSION

- 10 A.M. Address of Welcome
Response by President
Committee Reports
Paper: Using By-Products in Flour Mill Heating and Humidifying
—E. K. Campbell

AFTERNOON—HEATING SESSION

- 2 P.M. Paper: Determining Dry Return Proportions—R. V. Frost
Paper: Performance of a Warm Air Furnace with Anthracite and Bituminous Coal—A. P. Kratz
Paper: Selecting Wall Stacks Scientifically for Gravity Warm Air Heating Systems—V. S. Day
4 P.M. Automobile Trip

EVENING

- 7 P.M. Informal Reception and Dinner-Dance, Mission Hill Golf Club.

June 11, 1924

MORNING—RESEARCH SESSION

- 9 A.M. Paper: The Commercial Value of Heating and Ventilating Research to the Engineer—F. Paul Anderson
Paper: A Simple Method of Using the Katathermometer to Show the Effect of Air Motion on Human Beings—Margaret Ingels
Paper: Effective Temperature Applied to Industrial Ventilation Problems—C. P. Yagloglou and W. E. Miller
Paper: A Study of the Correlation of Skin Temperature to Physiological Reactions—W. J. McConnell and C. P. Yagloglou
Paper: The Heat Given Up by the Human Body and Its Effect on Heating and Ventilating Problems—C. P. Yagloglou
Paper: Some Additional Information Concerning Air Leakage around Window Openings—C. C. Schrader
Paper: The Flow of Steam and Condensation as Affected by High Pressures, Horizontal Offsets and Valves—Louis Ebin and R. L. Lincoln
Paper: Practical Applications of the Heat Flow Meter—P. Nicholls

2 P.M. **RESEARCH SESSION (continued)**

- Code of Minimum Requirements for the Heating and Ventilation of Buildings—Discussion led by—L. A. Harding, *General Chairman*
2.30 P.M. Golf Meet, Meadow Lake Golf Club. Inspection trip to notable Engineering Projects

EVENING

- 8 P.M. Banquet and Entertainment, Ball Room, Hotel Muehlebach

June 12, 1924

MORNING—CODE SESSION (continued)

- 9 A.M. Report of Committee on Minimum Requirements for the Heating and Ventilation of Buildings—L. A. Harding, *General Chairman*

AFTERNOON—VENTILATION SESSION

- Paper: Ozone and Its Use in Ventilation—Frank E. Hartman
2 P.M. Paper: Modern Trend in the Science of Ventilation—Perry West
Paper: Some Comments on Present Day Heating and Ventilating—W. S. Timmis

USING BY-PRODUCTS IN FLOUR MILL HEATING AND HUMIDIFYING

By E. K. CAMPBELL, KANSAS CITY, MO.

MEMBER

FLOUR is the world's best known food. Its use is universal but nowhere has production of this foodstuff been developed to such a high degree as in the United States. The history of the progress of the milling industry is amazing and a remarkable record has been made in producing a quality product. The heating and ventilating engineer has been consulted and has done his share in solving heating, humidity, ventilating and dust problems that have accompanied the advancement of flour milling practice.

A problem that has been of great interest to millers is that of by-product heat and moisture. Therefore, in 1922, when the Kansas Flour Mills Co., decided to erect a 6000 barrel flour mill in North Kansas City the writer was called upon to study the heating and ventilating problems. The company already operated 11 other mills with a combined capacity of 15,450 barrels of flour per day which represents the grinding of 69,525 bushels of wheat in 24 hours, figuring $4\frac{1}{2}$ bushels of wheat to the barrel of flour.

While the new mill was to provide for 6000 barrels capacity, machinery for only 3000 barrels was installed and is now in operation. The results noted in this paper are therefore based on the by-products from the production of only 3000 barrels of flour per day of 24 hours, and the effect on the building of double that capacity can only be estimated.

This building, Fig. 1, 240 ft. long, 65 ft. wide and eight stories high, contains about 1,500,000 cu. ft. of space of which 850,000 cu. ft. are to be heated. The estimated heat loss for the heated portion is about 3,000,000 B.t.u. per hour. The mill is of reinforced concrete, slip form construction and is what is known as a day-light mill being lighted by a large area of glass in steel factory sash. The machinery is electrically driven requiring motors of 1210 hp., the current for which is brought from the outside as there is no power generating plant provided.

Out of the experience in operating the other mills the Kansas Flour Mills Co., knew that a large amount of by-product heat is generated in the grinding process. Along with this heat is a large amount of moisture thrown off from the grinders as a result of treating the grain with water. This soaking of the wheat is done to keep the fine particles from flying which would mean loss of flour and an increase of the dust problem. These things are well known in the milling trade but little attempt

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Kansas City, Mo., June 1924.

has been made to save these by-products and use them although a mill must be both heated and humidified.

The Hoffman brothers of the Kansas Flour Mills Co., however, determined to make a serious effort along this line and after canvassing the possibilities, the writer's ideas were adopted as the basis, and he was authorized to put them into effect. It is not necessary to review the conferences by which we determined the results required. The writer knew nothing of the milling business and stated frankly that the owners must specify their requirements. After several talks a general outline was made about as follows:

1. It is necessary to heat the third floor of the mill to 60 deg. and all floors above the third floor to 70 deg. whether the machinery is in operation or not, no heat at all being provided for the first and second floors.
2. It is necessary to humidify the mill while in operation and provide approximately 65 per cent humidity summer and winter.
3. It is necessary to ventilate the mill in the summer while operating, to get rid of the surplus heat from the grinding process.
4. The owners desire to use this by-product heat and moisture which bothers them in the summer, to heat and humidify the mill when it is in operation in the winter.

This is not the order in which the problems came up. The owners had no definite outline in mind, and it took considerable questioning to get things in shape to know exactly what the problems were. They were, however, greatly interested in saving the surplus heat in the winter and applying it on the heating of the mill, thereby reducing the fuel cost. The outline above, however, is the logical order from the engineer's standpoint and will be the order of development in this paper.

Surveying this problem as a whole, it is apparent that a direct steam system will meet only the first two problems unless combined with a separate ventilating system which would run up the cost and still leave the problem of saving the by-product heat in which the owners were most interested unsolved. It seemed, therefore, that the situation demanded a furnace blast system for the following reasons:

1. If the waste heat was really to be saved a sectional plant in which the individual units could be quickly stopped or started was essential.
2. Since the mill is buying its power from outside, it needed the most economical heat generators—and steam boilers are not so economical as rightly built hot-air furnaces.
3. Large volumes of air must be moved both for summer ventilating and to save the waste heat in the winter.

Therefore a fan furnace system with special features was proposed and installed. The mill was started and our plant put in operation on August 21, 1923.

Taking up the problems as outlined it was necessary first of all, to heat the mill to 70 deg. in the coldest weather. On the basis of 70 deg. at 10 deg. below zero, the heat loss of the mill is approximately 3,000,000 B.t.u. per hour. To supply this loss four large furnaces were installed as shown in Fig. 2. These furnaces have been developed particularly for industrial work and the fire box is built of $\frac{5}{16}$ locomotive fire-box steel of the highest tensile strength and protected inside by the heat-pot of fire-clay tile just as in a smaller steel furnace. The joints are all electrically welded, no rivets being used. The flame and gases of combustion pass into the second chamber of radiating drum which is also welded. This drum con-

tains 21 vertical tubes each 8 in. in diameter and 5 ft. long, making a total heating surface for the furnace of 452 sq. ft. When erected each furnace stands 6 ft. high, 42 in. wide and 14 ft. long. These furnaces are not rated as boilers and furnaces are ordinarily rated but for our own guidance, each was figured to deliver 1,000,000 B.t.u. per hour without overheating the metal. This provides a safety margin of $33\frac{1}{4}$ per cent of the estimated requirement.

Referring to Fig. 2, it will be noticed that the smoke breeching is connected at the bottom of the radiating drum without by-pass or direct draft. A natural settling process goes on there at all times, the coolest gases are always at the bottom, the hottest gases remaining at the top falling toward the smoke outlet as they give up their heat and become heavier. The large amount of surface together with the large volume of air striking it combined with this settling process results in a very low flue gas temperature and in retarding the hot gases in the large com-

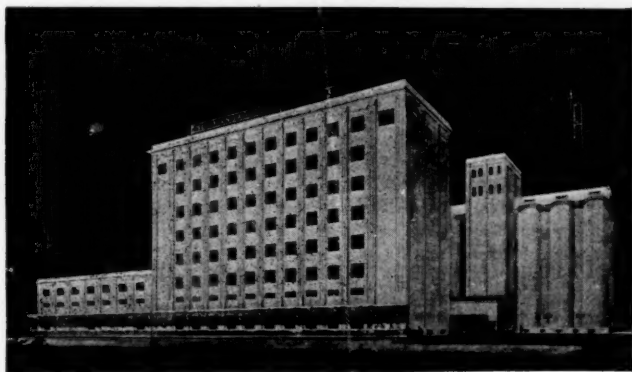


FIG. 1. A KANSAS CITY FLOUR MILL WITH 6000 BARREL CAPACITY

bustion chamber until there is good mixture and the combustion process has reached a high percentage of completion. A small amount of secondary air is discharged above and at the front of the fire to aid this process. It is taken in through the draft door so no special control of this secondary air is needed.

A question frequently asked is, "How does this affect the draft?" We have never been able to discover any effect and do not as a rule require as strong flues as are specified for ordinary steam practice.

Four of these furnaces comprise the heating plant. They are all set in one brick enclosure as shown in Fig. 3, so that one or more can be fired as occasion requires, giving the sectional feature necessary to apply the waste heat to the reduction of fuel consumption and to fit mild and changing weather conditions.

The air is moved over these furnaces by two 8 ft. Parker-Hope fans which have blades in the shape of a half globe, a modification of the cone type and are very efficient low pressure volume blowers. Each fan is belted to a 10 hp. motor with reducing pulleys so that each fan is operated at about 150 r.p.m. at which speed and under the conditions which prevail in this particular installation, they are delivering about 100,000 cu. ft. of air per min., giving a change of air once every 8 min.

Before we finish experimenting in the operation of this plant this speed will be

increased as the motors now show only 52 per cent load and use but 4.08 kw. of current each. This will be particularly for summer use as this speed and volume are satisfactory for winter purposes.

The duct system is shown in Fig. 4, being a cross section of the mill showing the duct rising and discharging to the various rooms. The duct system is laid out on a recirculating plan, one duct on the right side being return air duct, one on the left the warm air or supply duct. The fan will therefore draw the cold air off each floor through openings located close to the floor, pass that air over the furnaces, driving it through the supply duct into the various rooms. All openings are dampered so that the temperature can be controlled by referring to thermometers hung at various points in the building. No attempt is made to supply automatic regulation.

Because the system is laid out on the recirculation plan there is no necessity of using distributing ducts to drive the heat to the various parts of the different floors, and although the main room on each floor is 111 ft. long with both the supply and return openings at the same end, there is no difficulty in getting an even distribution of the heat throughout the room without distributing ducts. This principle has been successfully applied in rooms as long as 780 ft. and we have found no occasion when this scheme is properly applied, where it failed to get as perfect a distribution as can be desired. It is the writer's belief that if those engineers, who at the present time are worrying so much about problems of distribution, would lay aside their prejudice against recirculation, they would find that many of their problems would automatically disappear without producing new problems to anything like the extent they fear. Neither does the writer believe that ozone is necessary as a purifier, except as a deodorizer.

It will be noticed from Fig. 4 that the ducts as they rise through the mill above the third floor are in waste space between the flour bins. Through the first and second floors they occupy space which might otherwise have been used, but above the third floor they occupy no space that would have been of any value.

This simple installation is designed to and the writer thinks will, without question, heat the mill to 70 deg. in the coldest weather that comes in this climate without the aid of the by-product heat. Also we think it will heat the mill to 120 deg. when it is 90 deg. outside. Mill men state that this is a very important matter. There are a number of disinfectants which can be used, each one of which will exterminate some of the pests but not the others and the only thing that will get them all is to heat the mill to 120 deg. continuously for 24 hours. (Some millers think as high as 150 deg. is needed, but the mill being considered specified 120 deg.) That, they say, reaches all of the pests and destroys not only the pests but any eggs that may have been brought in with the wheat. Of course the mill is shut down and closed tight at such a time.

After considerable discussion about the problem of humidification, the use of air washers and other means of moistening the air, the fact was finally brought out that they wanted us to include in our work a method of heating 700 gal. of water per hour which they use to treat the grain and get it in proper condition to go to the grinding machine. In the various processes which follow the grinding this moisture is, of course, gradually evaporated and escapes into the mill, being carried by the air of the conveyor system and discharged to the eighth floor. With this amount of moisture discharged into the mill every hour it is quite evident that the humidification problem will take care of itself as long as this moisture is recirculated by the heating system. There would be, therefore, no humidifying problem when both the mill and the heating plant are in operation.

This proved to be the case in operation, and the mill was humidified to about 60 per cent during the fall months. But trouble resulted in the colder weather on the eighth floor where of course the moisture is very excessive and where the thin concrete walls (about 8 in. thick) cause great condensation. Various makeshifts were tried during the winter to overcome this trouble but, of course, without success. Only thorough insulation of the walls will correct this condition and this is to be done during the coming summer.

This of course caused the upsetting of the main feature of the system in severe weather to a considerable extent, so that results in both humidification and fuel saving were not up to expectations or possibilities in cold weather. Yet in spite of this condition, two of the four furnaces together with the by-product heat proved enough to heat the mill when the temperature outside was 15 deg. below zero.

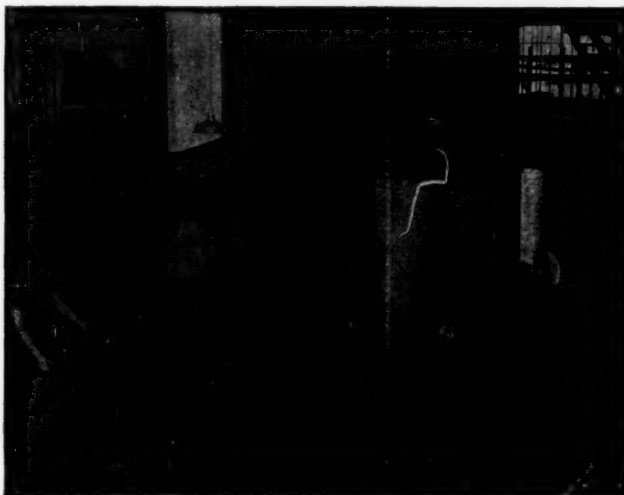


FIG. 2. FURNACES FOR INDUSTRIAL HEATING—FOUR OF THESE UNITS USED IN KANSAS CITY FLOUR MILLS PLANT

When the mill is not in operation there is, of course, a humidifying problem because it is necessary to maintain the proper percentage of humidity to protect whatever stock is in the mill and also to keep the air in proper condition for the men at work. Also, when the mill is in operation in the summer time, and the heated air from the dust collectors carrying its moisture is discharged from the mill, there will then be a humidifying problem which must be taken care of. New air will, of course, be drawn into the mill continually and this air must be humidified so that the humidity of the mill is maintained at about 60 per cent. This problem, will of course, be more acute in dry seasons and the great need of humidification will be during the months of July and August, or during a season when the rainfall is scant and the air generally is dry.

To supply this humidity spray heads have been installed in the rising column of air using compressed air for atomization. While no attempt was made at automatic

control, these sprays can be controlled by the head miller to meet any set of conditions. A similar set of sprays in the Sand Springs Cotton Mills is humidifying the weaving room to 95 per cent in the winter months.

The problems presented by the generation of heat in the grinding process are exceedingly indefinite. It appears that no one knows how much heat is generated in the grinding of a bushel of grain, or any other quantity. It also appears that the quantity is not the same any two times. It varies with the quality of the grain. It varies with the preparation of the grain. It varies with the speed of the milling process. It varies with the conditions of the air in the building. It is an extremely variable and indefinite quantity yet the millers all agree that it is a very considerable amount and worth attempting to save.

It comes about, of course, just in the same way that the head of a wooden stake gets hot if you hammer it with a mallet. In the modern mills, it is a repetition of the process of crushing. Each process doing just as little as possible. This is the distinction between the gradual reduction mill and the old time mill which simply ground the flour between a pair of burrs. In this grinding or crushing process, the wheat itself becomes heated, the flour comes out of the mill heated, the bran is heated, the shorts are heated, the rolls themselves are heated and the air which is used in the conveyor system is heated.

Any attempt to trace the travel of the heat through these various items is a difficult problem. Some of it, of course, goes with the flour, with the bran, with the shorts, into the packing room and is lost to the mill. All of this material is delivered to the packers at about 90 deg. That which goes into the rolls themselves is transferred by conduction into the balance of the machine, and from them by convection into the air of the room in which the machine stands. Thus sometimes the roll floors become very hot although in most of the modern mills the roll floors are not excessively heated and most of the heat is transferred by convection to the air of the conveyors system and discharged into the mill or outside as the case may be through the dust collectors.

In this particular mill, there are 30 Niagara cloth tube dust collectors which discharge the air into the seventh and eighth floors so that the heat thus discharged into the seventh floor will rise promptly into the eighth. These dust collectors are supposed to take all of the dust out of the air. Of course they are not 100 per cent efficient, but the millers assure us that they are near enough 100 per cent efficient so that we would not have to worry about the dust problem and no air washers or other means of cleaning the air were required in connection with the heating system. Experience in this regard is somewhat indefinite as yet.

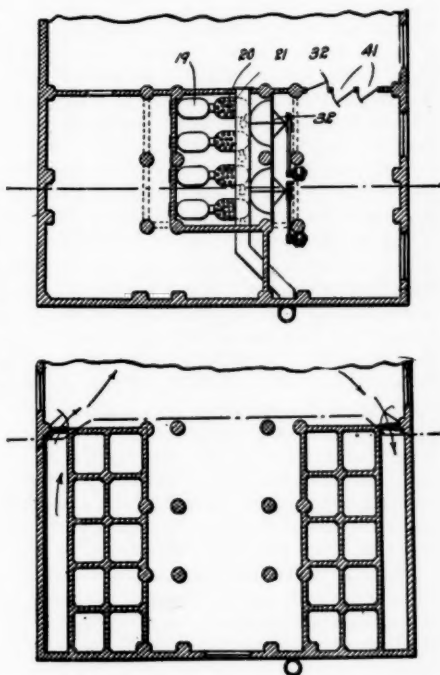
This type of dust collector discharges all of the air of the conveyor system and therefore the heat which is generated in the grinding process inside of the mill except the heat which is carried out in the warm finished products. This fact creates two problems that are both highly important to the millers, and in which the Kansas Flour Mills executives were particularly interested, one is the expelling of this heat in the summer time, and the other is saving the heat in the winter.

The ventilating problem in the summer is accomplished purely by the volume of cooler air driven into the mill and, of course, creating a pressure within the mill just the same as in any plenum fan system. In the roof above the eighth floor are very large ventilator openings under control. They will be closed in the winter, wide open in the summer, so that the pressure created by the fans will drive out the hottest air of the mill, which will hug the ceiling of the eighth floor, and go imme-

diately out through these ventilator openings. This will drive out a large part of the heat directly. Dampers can be controlled on the various openings so that the pressure can be directed against this top floor or against other floors should they show any tendency to accumulate surplus heat.

You will notice on the basement floor plan, Fig. 3, three large doors are shown between the basement and the air chamber immediately behind the fans. There are two windows opening directly into the air chamber, and a number of other windows opening into the general basement. These windows are all shaded by the

FIG. 3. SHOWING SOUTH END OF FLOOR PLAN FOR BASEMENT AND TYPICAL FLOOR. BASEMENT PLANS SHOW FOUR FURNACES AND TWO TO EIGHT FANS IN BRICK ENCLOSURE. NOTICE DUCTS ARE IN WASTE SPACE BEHIND FLOUR BINS



concrete loading dock. There are a number of these openings on both sides of the building so that air can be drawn from the coolest places surrounding the building, aiding further the summer ventilating by getting the coolest air possible.

With the fans in operation all night, as the mill is a 24 hour mill, the building will become thoroughly cooled each night and consequently can be kept comfortable by the moving of the air even on very hot days.

The cooling of the mill and driving out of this surplus heat is an important matter from the standpoint of the production of the flour, and we know of one case in which a mill with no provision for ventilation increased the production of the flour 100 barrels per month by installing exhaust fans to take the heat away from the grinding flour in the hot weather.

The volume of 100,000 cu. ft. of air per minute, which has been mentioned, can be increased to approximately 140,000 by speeding the fans to about 220 r.p.m. at which speed the motors would approach full load. A much larger volume of air would be required to take care of the summer heat were it not that the fans operate all night as well as all day, during which time they get air which thoroughly cools the walls as well as the air of the mill, preparatory to starting through the hot part of each day.

The reclaiming of the surplus heat has now become a very simple matter. The foundation for it has been laid in the general layout of the system, and the only change which needs to be made to meet this problem is to carry the return air duct full size clear to the ceiling of the eighth floor instead of reducing it at each floor, as would be the natural thing to do if it were a straight heating plant. It will be noted in Fig. 4 that the return air duct maintains full size to within about 3 ft. of the roof, as the hottest air of the mill will be next to the roof under conditions existing in this mill. This simply means that the fans at the bottom of the return air shaft will be continually skimming this heated air from the top floor, pulling it down through the return duct, over the furnaces, which will be fired or not as needed, and then driving it into the distributing ducts, discharging to the various floors as may be required just as if the heated air was all coming from the furnaces.

The conditions that we have to meet then are three fold. At times all of the heat will be generated by the furnaces, at other times there will be no heat at all passed through the fans, and at other times, when the mill is in operation in the winter, all of the surplus heat which can be reclaimed will be passed over the furnace and through the fans, supplemented by one or more of the furnaces in service as the need develops. It was our hope, and it proved out, that in some weather the reclaimed heat alone would be sufficient to warm the building.

Of course the amount of heat that would be saved by such process was and is problematical. The amount could only be estimated in an indefinite sort of way from the rule which is generally accepted as approximately correct among the millers, that about 40 per cent of the horse power used in driving the mill is used in the actual grinding process and is turned into heat. The other 60 per cent is used in the conveyor system and in the friction of machinery itself which, of course, generates heat, but a very negligible amount.

This mill has a total motor capacity of 1210 hp. for the machinery now installed. Of this, however, only 1075 is used in the operation of the mill. Forty per cent of this is 430 hp. Assuming that this entire 430 hp. is turned into heat, at 2546 B.t.u. per horse power, this will figure 1,094,000 B.t.u. per hour. This is slightly more than one third the estimated requirement of the building at 10 deg. below zero.

This was advance figuring and no one knew how near the mark it might hit but it proved to be substantially accurate. Without any fire in the furnaces the by-product heat maintained a temperature of 70 deg. when it was 40 deg. outside, actually supplying three eighths of the total requirements, or about 1,250,000 B.t.u. per hour. When the machinery and the by-product heat are doubled it is hoped to double the fuel saving though it may be necessary to speed up the fans to maintain bearable conditions on the top floor.

The total fuel saved for the winter is not what it should have been because much fuel was wasted in experiments carried on by the miller trying to solve the sweating problem on the top floor. The total fuel used for the winter was 1150 barrels of fuel oil. An interesting comparison has come to the writer from the Midland Mill

also in North Kansas City. It is a 1700 barrel mill with estimated heat loss of about 1,250,000 B.t.u. per hour. It is heated with direct steam and air, and moisture from the dust collectors is discharged to the atmosphere. Three thousand barrels of oil were required to heat it and this has been the average for several

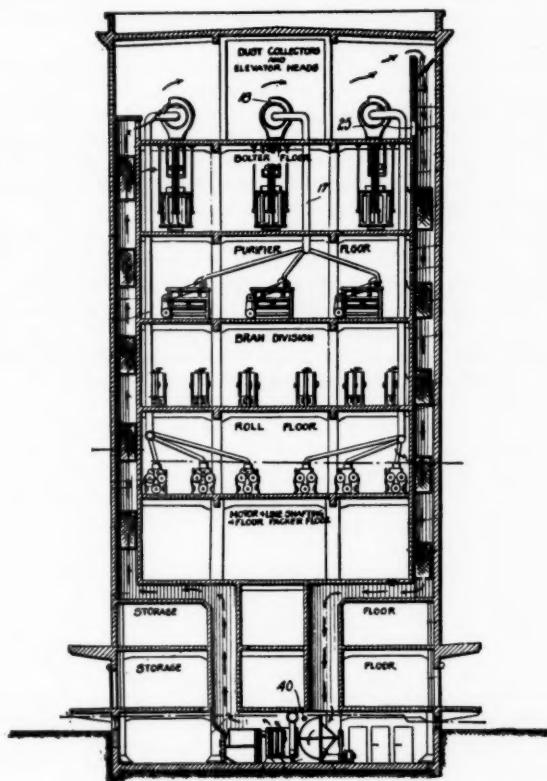


FIG. 4. SECTION THROUGH DUCTS—FURNACES AND FANS

years—ever since the mill was built—nearly three times as much fuel to supply less than half the heat losses.

Reviewing the results obtained and the problems met, these facts seem to be established:

1. The by-product heat was very evidently underestimated.
2. The system seems to make available for use in heating a building all of the by-product heat except that which is carried out in the finished products.

3. Proper operating conditions for such a system with the air discharged free from the dust collector into the top floor requires thorough insulation to prevent sweating. This might be obviated by taking the air direct from metal dust collectors to the fan without discharging it into the top floor.

4. There seems to be ample humidity provided with the exception of the hot weather when the heat and humidity from the dust collectors are discharged from the mill.

5. The actual saving in fuel under proper operating conditions, is about thirty per cent of the total requirements of the building in this particular mill where there is only half of the machinery installed. When additional machinery is in operation I believe this estimate can safely be raised to 55 per cent. This percentage does not include the saving due to the use of the furnace fan system instead of direct steam, which the Midland Mill comparison indicates is very considerable.

6. The dust problem may prove more serious than was anticipated. That also could be handled by taking the air direct from a metal dust collector and not discharging it into the mill until it had been cleaned.

As a whole the experiment has justified itself. It has paid its way and when carried to final conclusion under the best operating conditions, will prove a real method of saving fuel for the millers and so far there seems to be no obstacle which cannot be successfully and fully overcome.

DETERMINING DRY RETURN PROPORTIONS

By R. V. FROST, NORRISTOWN, PA.

MEMBER

Experimental work that develops factors for practical application is of immeasurable value to the practicing engineer, and is a decidedly worthwhile contribution toward the progress of the profession. A recent example of such work in our field is that of Prof. L. S. O'Bannon on The Simultaneous Flow of Air and Water in Pipes, so widely commended at the last Annual Meeting. The results are of unusual practical application and give us an opportunity to analyze the factors governing flow of water and air in return lines in a manner heretofore impossible. It is the writer's pleasure to be able in this paper, to present to the Society an analysis of these factors, and to show how they must be taken into account in properly proportioning dry return capacities.

Chart Fig. 1 is the complete O'Bannon chart for simultaneous flow of water and air in 1 in. pipe, graded 1 in. in 10 ft., while chart Fig. 2 is that section of the O'Bannon chart that covers the field of practical application.

Instead of giving capacities for water flow in pounds of water per hour, Fig. 2 shows corresponding values in square feet of radiation: thus, the 100 lb. line on the O'Bannon chart is the 400 sq. ft. of radiation line in Fig. 2.

Since each vertical line in Fig. 1 represents the carrying capacity of the pipe for a certain depth of flow at a given grade, it has been possible to establish corresponding carrying capacities for the same depth of flow at other grades; as shown by the values set opposite the grades of 1 in. in 20 ft., 1 in 30 and *vertical*. In this particular, however, it must be remembered that the O'Bannon experiments have not been completed, and consequently, other grade conditions than 1 in 10 must be based on calculation from data of other authorities.

On the vertical side of Fig. 2, in addition to the pressure drop for 100 ft. of length, corresponding pressure drops for multiple lengths of 100 ft. have been shown.

In working out the capacities and resistances in return pipes from these charts it has been necessary to establish certain basic values; the first, being the steam space volume in a direct radiator. This volume, without checking by actual measurement, has been calculated at 3 cu. ft. per 100 sq. ft. of radiation. Having established this value and setting some time limit for the ejection of air from the radiator, it is a simple problem to determine the relationship between capacity and pressure drop, or back pressure.

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For example: If the air in a 400 sq. ft. radiator is ejected in 10 min., the rate of ejection will be 1.2 cu. ft. of air per minute. This point being plotted on the chart as shown.

In a similar manner, other points are plotted until the set of curves marked as the "Pressure Drop—Capacity Lines," have been established. By means of these curves, it is easy to read the pressure drop in the line or the back pressure, or pressure resistance,—as may be the preferred term, for any given relationship of capacity, time allowance for air ejection, and size and grade of pipe.

Having prepared the chart, the conditions entering into the proportioning of the return line can be studied. Assume that we have a capacity load of 300 sq. ft. of radiation for a 1 in. dry return line. Fig. 2 shows capacities of 280 sq. ft. for a grade of 1 in. in 20 ft., 300 sq. ft. for a grade of 1 in. in 10 ft., and 280 sq. ft. for a vertical pipe. Since under each of these conditions, the same volume of water is passing through the pipe in a unit of time, and also the same volume of air, the rate of flow of both the water and air is affected by the change of grade, and therefore, the pressure drop or resistance to the air flow decreases in proportion to the increase in grade. The change in resistance, however, is very, very slight and can be ignored, as the total resistance itself is a very small factor, being approximately 0.1 oz. per 100 ft. of pipe.

However, if the load on the line is increased to 400 sq. ft., it is discovered that the resistance has taken a jump to about 0.18 oz., an increase of 80 per cent as compared to but 33.3 per cent increase in capacity, and if the capacity be further increased to 600 sq. ft., the pressure drop jumps to 0.45 oz., an increase of 450 per cent in resistance as compared to 200 per cent increase in capacity.

If a longer time allowance is given for the ejection of the air in the radiator, a 1 in. pipe, graded 1 in. in 10 ft., will carry 600 sq. ft. of radiation with a pressure drop of less than 0.1 oz. per 100 ft., for a 30 min. interval, and that 1000 sq. ft. of radiation can be carried with a pressure of not to exceed 0.2 oz. If the opposite direction is taken and an attempt made to eject the air in 3 min., then it is found that for 300 sq. ft. of radiation, the pressure drop increases to 0.8 oz., and for 400 sq. ft., to 1.5 oz.

It is well to note that all of these points are far below the *border line for region of steady flow*; that is, any of the capacities that have been considered can be safely carried in a 1 in. pipe without danger of a turbulent or noisy flow.

In the case of air flow, it is apparent that the only effect to be considered is that with the increase in capacity and resultant increase in resistance to air flow, a point is reached at which the resistance in the return line balances the steam pressure in the radiator, and further increase in capacity causes an increase in the time for air ejection.

The increase in resistance to air flow has no effect whatever upon the flow of condensation.

The length of the return line can have no effect upon the flow of condensation, owing to the impulse to flow being caused by the grade of the line, but it does cause an increase in resistance to the flow of air, in direct proportion to the increase in length, with the result that a longer interval of time is required to eject the air in the radiator. The principal advantage to be gained by making a return line ample in size is to reduce to a minimum the back pressure on the radiator, thus causing a free flow of steam to the most distant radiator, and provided the steam supply lines are properly balanced, bring about a uniform heating of all radiators.

It is important, in the study of this problem that one maintain the proper perspective in order not to go to extremes. It is the writer's own inclination to lay great stress on the grade of the pipe, while others go to the other extreme in laying too great stress on the pressure drop; whereas, an examination of the chart shows that the same capacity can be carried in the same size pipe on a line in which the grade varies from 1 in. in 30 ft. to a vertical, without seriously affecting the pressure drop, and likewise, owing to the very low pressure drop under usual conditions, if the size of the line is changed to allow for increased length, the line becomes unbalanced to a greater extent than if one uniform size for all reasonable lengths is maintained.

The important problem to be solved by the aid of these charts is to select the ideal carrying capacity for each size of pipe. At present, the incomplete condition of the

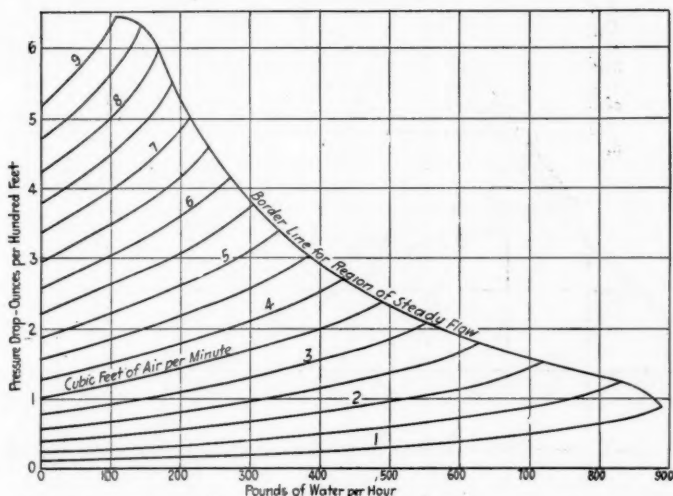


FIG. 1. A CAPACITY CHART FOR SIMULTANEOUS FLOW OF WATER AND AIR IN ONE INCH PIPE—PARALLEL FLOW

experimental work prohibits our going beyond the one size given on this chart; nevertheless, it is possible to illustrate very thoroughly, the influence of the contributing factors upon the carrying capacity.

First: It must be borne in mind that any capacity rating given to a pipe is based on ideal conditions, and that sufficient leeway must be allowed for abnormal conditions, such as a cold radiator and a cold room, which would tend to increase the condensation rate above normal. It is important that the capacity be kept well below the line for unsteady flow.

Second: The pressure resistance to flow must be maintained low, to permit uniform circulation, to avoid increased boiler pressure and to permit rapid ejection of air from the radiator.

As an example, suppose standard practice for a vapor or atmospheric system of circulation is selected. Assume 400 sq. ft. capacity as the rating for a 1 in. dry

return line for this class of work. An analysis of the O'Bannon chart shows that if the air in the radiator is ejected in 30 min., the resistance is 0.05 oz. per 100 ft. of pipe, if it is ejected in 20 min. the resistance is 0.1 oz., and if ejected in 10 min. the resistance is 0.2 oz. If the line is 400 ft. long, the resistances become 0.2 oz., 0.4 oz., and 0.8 oz., respectively. If an attempt is made to balance the capacities for the 400 ft. line as compared to the 100 ft., then the capacity must be reduced to less than 200 sq. ft. on the 400 ft. line in order to maintain the same pressure drop as for the 400 sq. ft. on the 100 ft. line. The resistances given are for the capacity of 400 sq. ft. in the pipe graded 1 in. in 10 ft. If the grade is decreased to 1 in 20 or 1 in 30 then the resistance is increased, but the proportional increase is so slight that the increased resistance amounts to but 0.025 oz. for the decrease in grade from 1 in. in 10 ft. to 1 in. in 30 ft.

In reference to the pressure drop in modulation systems, one well-known authority makes the following statement:

Pressure drop in the return main is variable—greatest during initial heating and dependent on length of run and maximum velocity. In a well-proportioned system, the pressure drop should never exceed 0.05 lb. per sq. in. differential between the farthest radiator trap and the vent, and during normal heating is so slight as to be almost negligible.

In this quotation, a basic fact is so well stated, that the writer proposes to make this the basis for establishing the capacity of the dry return lines for this class of service. The 0.05 lb. pressure drop quoted—equivalent to 0.08 oz., is the maximum resistance that we have allowed for a line of 1 in. pipe, 400 ft. long, carrying 400 sq. ft. of radiation.

Therefore, it is the writer's proposal that it be established as standard practice for modulation, vapor or atmospheric systems using radiator traps: *namely*, that a 1 in. pipe, graded not less than 1 in. in 30 ft., and not exceeding 400 ft. in length, including allowance for turns in direction, be rated to carry the condensation and air from 400 sq. ft. of radiation gross, and that 10 min. be the maximum time interval for ejection of air from the radiator, or time for the radiator to heat completely.

Having established this basic standard, all pipe sizes can be rated in ratio to the above.

If the conditions for a vacuum line are maintained as stated: that is, the rating based on 400 ft. in length, and the time interval for air ejection set at 10 min., then there is a pressure drop of 2 oz. per 400 ft., on 600 sq. ft. capacity in the 1 in. pipe. The conditions under which a vacuum operates vary from those of a modulation system, in that as soon as the air is ejected from the radiator, then re-evaporation of the water of condensation forms a considerable volume of vapor, which must be taken into account. If the evaporation is based at $2\frac{1}{2}$ per cent and the volume of vapor at 40 cu. ft. per lb., the resultant volume of vapor per minute is equivalent to the volume of air to be ejected per minute.

On this basis the rating for a 1 in. vacuum line, not to exceed 400 ft. in length, graded not less than 1 in. in 30 ft., would be rated at 600 sq. ft. If the line is increased to 800 ft. in length, the capacity must be decreased to 440 sq. ft. in order that the pressure drop does not exceed 2 oz. maximum. In establishing this rating the maximum allowable pressure drop was arbitrarily set at 2 oz. and on comparison the rating was found to agree very closely with what may be called standard practice.

All the foregoing discussion has been based on the parallel flow of water and air.

Counter flow was also covered by the O'Bannon experiments, but an extended discussion is unnecessary as within the area covered by practical return line proportions, the relationship between capacity and pressure drop differs but slightly from that on parallel flow.

In establishing standard tables for pipe sizes it is important that the view point of the designer be constantly kept in mind, for details can easily be added that would serve to confuse rather than assist the user. To obviate this possibility, let our return line tables be reduced to one capacity for each size of pipe with a statement of the limiting conditions under which the rating was established. In this way, systems can be as well balanced as is possible with the commercial sizes of pipe, as can be done by the aid of some of our present super-refined tables.

In this connection it is suggested that the sections on Dry Returns of Tables 27, 28 and 29 of *THE GUIDE*, 1923 be changed to contain but one column with the heading "Dry Returns not exceeding 400 ft. in length graded not less than 1 in. in 30 ft.," and that the values be those given in the present column for Vertical Pipes. Further that Table 31 be discarded as containing too much detail and Table 36 must stand substantially as given.

The SOCIETY is indeed indebted to Prof. O'Bannon for his exhaustive research work that has placed such a fund of information on this subject at our disposal, and it is to be hoped that his work will be continued to its final completion.

OZONE AND ITS USE IN VENTILATION

By FRANK E. HARTMAN, PITTSBURGH, PA.

MEMBER

The Constitution of Ozone

OZONE is an allotropic modification of oxygen. Allotropy is a term used to describe the existence of elements or compounds, while possessing the same chemical composition, display different physical or chemical properties. Carbon presents a well known example of allotropy. All of us know carbon in three distinct forms; the diamond, graphite, and the amorphous varieties such as lampblack, etc. Everyone of these modifications of carbon possess the same molecular weight and are composed solely of carbon, yet they display important chemical differences, while their physical differences are quite palpable. A further excellent example is found in phosphorus. Red phosphorus, such as is used for matches, is quite inert towards the atmosphere at ordinary temperatures, while the yellow, or normal, variety reacts with explosive violence with the oxygen of the air at ordinary temperatures. Both of these modifications are composed solely of phosphorus, and further possess the same molecular weight.

These examples serve to illustrate the enormous differences in the behavior of chemical elements when existing as allotropic modifications, and they have been cited, primarily, to show that ozone is in no sense unique, but displays chemical differences from oxygen that are readily ascribable to a state of allotropy.

Considerable trade literature on ozone that I have read tends to treat this substance as though it were one of the baffling mysteries of present day science. Such phrases as—"Ozone is generally conceded by chemists to be a form of oxygen, "Ozone is undoubtedly an electrified oxygen"—occur and reoccur all through these little advertising brochures, and their general tone is one that tends to infer that ozone is an intangible substance, something like our hypothetical aether, that is not susceptible to the broad generalizations of applied science. To better drive this point home these little brochures avoid quantitative treatment of the subject as one would the plague. The natural inference that follows such procedure is, that quantitative treatment is not possible, or of such a complicated nature as to exclude it from the scope of the engineer.

This is not good for any industry, and ozone today has undeniably taken its place as a stable industry. No one wants to invest one's clients' money in uncertain ventures, nor is the engineer given to buying pigs in pokes. Of course, we

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can at any time say that all of our knowledge is simply relative, that nothing is absolute. Practically any theory that serves science so ably today can be made ridiculous by pushing it out to its ultimate logical sequence. But such is the task of the abstract scientist, and even he dare not tear down an edifice of theory unless he has the materials from which he may construct a better one in its stead. The engineer is interested in the concrete, and while he may use abstract truths to guide his judgment, he can only employ the practical in the execution of the tasks entrusted to his care. Anything that is not susceptible to mathematical treatment can be of but little practical use to the engineer.

As before stated, ozone is an allotropic modification of oxygen. Exceedingly reasonable, and decidedly familiar laws govern its formation, and the technique of its quantitative determination is familiar to every routine analyst and is certainly a part of the mental equipment of every chemist or chemical engineer. It does not possess a molecular weight in common with oxygen since ozone is composed of three atoms of oxygen, whereas normal oxygen has but two atoms to the molecule. Oxygen is normally, and persistently, a diatomic element, hence, the addition of an extra atom strikes at molecular stability, therefore, ozone is highly unstable, and it is this instability that gives it its chief virtues.

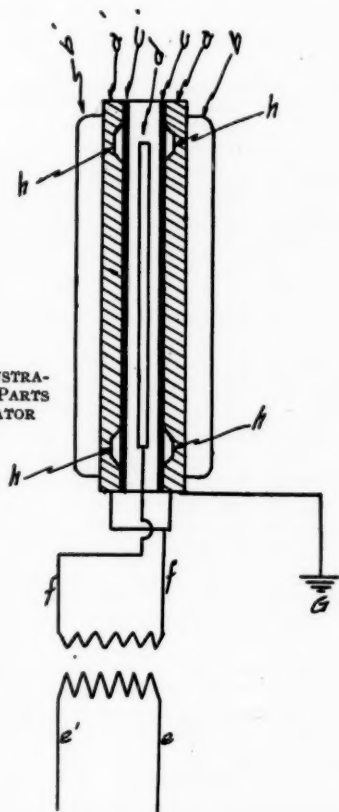
Production of Ozone

Commercially, ozone is produced through the agency of a high tension electric discharge. Fig. 1 illustrated diagrammatically, the essentials of an ozone generator, *a* and *a'* are the outer electrodes; *b* and *b'* are radiation fins employed to maintain the electrodes cool; *c* and *c'* are pieces of dielectric materials and *d* is the central or inner electrode. The high tension electric potential is supplied from the step-up transformer *T*, of which *e* and *e'* are the primary supply wires; and *f* and *f'* the secondary wires carrying the high tension current to the generator. It will be noted that the outer electrodes are connected to the ground at *G*, thus the exposed parts of the generator are always of the same potential as the earth, and, in consequence, cannot impart an electric shock to any one handling the apparatus. The electric circuit through the generator is practically that of a condenser thrown across the line, offering a capacity resistance to the flow of the current. Now when such a circuit is energized, electric charges will be present on the surface of the electrodes and the current will flow from one electrode, through the dielectric, and across the air space to the other electrode. At any given instant the current flow may be regarded as constant, and there will be proceeding from the negative to the positive electrode a stream of electrons and their passage through the gas space will serve to ionize a part of the gases contained in the field of discharge. This phenomenon is difficult to describe in a few words, and as much has been taken for granted no attempt will be made to explain how the current starts to flow in the first place. For a moment consider the structure of an atom. At one time an atom was considered to be the smallest division of matter. If an atom were magnified many millions of times, however, we would see a system having a stationary nucleus around which infinite particles revolve in a high state of motion. These infinite particles are electrons, and are now considered to be the smallest possible division of matter. Ordinarily an atom is a system in perfect kinetic equilibrium, and its electrons disport themselves quite properly. But it is possible, by vibration, heat, light, or other forms of radiant energy, to drive off some of the electrons from the atom, or attach other electrons thereto. Now the attachment of an electron to a gas molecule produces the ionization here referred to. But the effect of the electron stream on oxygen is, according to Rideal,¹ three fold:

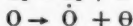
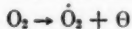
a. A splitting of the molecule into two neutral atoms by direct impact,
 $O_2 \rightarrow O + O$

It is thus that ozone is chiefly formed by ultra-violet light,
 $3O \rightarrow O_3$

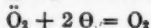
FIG. 1. DIAGRAM ILLUSTRATING THE ESSENTIAL PARTS OF AN OZONE GENERATOR



b. An ionization of the atom or molecule by impact,



The positive molecules so formed may be atomic or consist of molecular clusters (as there is evidence extant of clusters up to O_6). The atoms or molecules with one or two positive charges proceed in the reverse direction to the stream of electrons and by impact and combination with them, neutralization to atoms and molecular groups is once more effected.

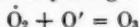


c. An ionization of the molecule or atom by impact and the adherence of an electron.

The electron having spent most of its kinetic energy with which it left the dielectric, may, on contact with a neutral atom or molecule, not possess sufficient energy to detach a valency electron, but actually adhere to the system and form a negatively charged atom or molecule:



Oppositely charged atoms and molecules may then react to form ozone:



The theory just outlined is well confirmed in the work of Puschin and Kauchtshev², and is further confirmed by the work of Starke³ and the author.⁴

Under the head of the production of ozone the external factors that have bearing on the efficiency of ozone production should be considered. The literature, especially that dealing with actual experiments on the production of ozone, is profuse in its reference to the necessity of carefully conditioning the air from which ozone is to be made. Many experimentors describe the most elaborate measures for drying and filtering the air served to their ozone generators, while none has been so careless as to neglect this factor entirely. There are several reasons why clean and reasonably dry air is essential to the best efficiency, constancy of output, and continuous operation of an ozonizer. Primarily, the brush discharge is very energetic in the precipitation of vapors and solid particles suspended in a gas, the mechanism of the precipitation being analogous to that of the Cottrell electrostatic dust precipitator. If the air contains dust or water vapors, a large portion of each will be precipitated by the discharge and lodged either on the electrode or the dielectric, while such as escape precipitation will carry off electric charges, thus decreasing the ozone output of the generator. The presence of dust on an electrode or dielectric appreciably changes the character of the discharge and seriously interferes with the ozone production. These changes are brought about through the tendency for dust particles to accumulate charges and thus cause a concentration of energy into a very small area. This concentration of energy is productive of high temperatures and thus the dielectric properties of the dielectric material undergo a change and greater current is permitted to pass. With the addition of more current the temperature rises rapidly and greatly augments the conductance of the gas. With augmented conductance more current is allowed to flow and this condition, in its acute form, is productive of the high tension arc, which almost immediately degenerated into the hot, low tension arc, which, if allowed to persist, would destroy the generator. In its milder form the temperature rise is not sufficient to rupture the dielectric, but is sufficient to exert an effect on the nitrogen contained in the air and oxides of nitrogen are produced. In the case of aqueous vapors the net result is the same but the mechanism is somewhat different. The precipitated water accumulates charges and thus concentrates the energy in the same way as the dust. In addition, however, the water absorbs the oxides of nitrogen and an electrolyte of nitric acid is formed on the dielectric. This acts as though a third electrode had been added to the system, and dielectric rupture occurs much more rapidly than in the case of the dust particle. The change in the character of the discharge, brought about by the presence of water or dust on the dielectrics, is further productive of changes in the ozonizing power of the discharge. In consequence, a constant output cannot be obtained from an ozone generator, unless the air supplied it has been properly conditioned. It is not nec-

essary to have absolutely dry air, for apparently there is a limit to the vapor precipitating power of the discharge. No ill effects are noted with air having a relative humidity of 10 to 15 per cent at a dry bulb temperature of 70 deg. fahr. but even a very small amount of dust will effect the discharge, therefore, the air should be as clean as possible. The necessity of properly conditioned air is well illustrated by the charts, Figs. 2 and 3.

Apparently there is no definite relation between ozone production, potential, and current density. The dielectric constant of the dielectric material, and the distance between the electrodes of a given generator, determines its optimum potential. When the air supplied the ozone generator has a constant water content, the ozone production is a definite function of the energy density.⁴ By the energy

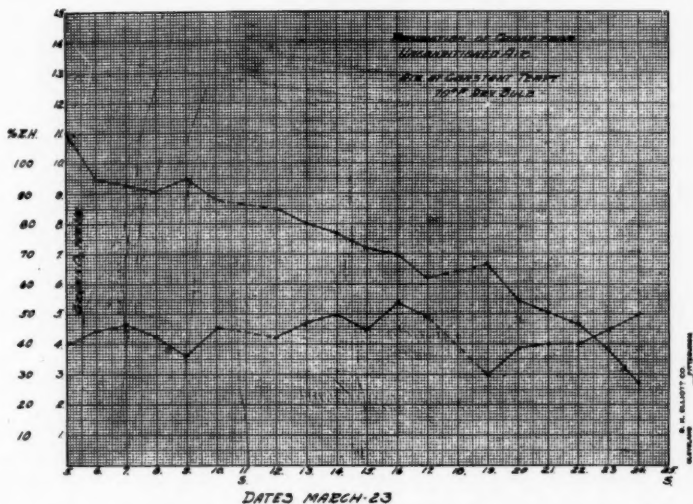


FIG. 2. CHART OF PRODUCTION OF OZONE FROM UNCONDITIONED AIR

density is meant the energy input to the ozonizer. This fact has been used in evolving an ozone meter.

Ozone equipment such as ours, using conditioned air only, the energy input can be measured with a wattmeter and by making a chemical determination of the ozone output over the entire range of the energy input, the scale of a wattmeter can be calibrated so that it will read in terms of ozone instead of watts. With such an instrument the most unskilled janitor can, at all times, know positively just the amount of ozone that the apparatus is producing, and properly phase it with the amount of air that is being circulated.

At concentrations above 1 gram of ozone per cubic meter of air the relation between the amount of ozone produced and the velocity of the air through the ozonizer is shown in Fig. 4. At concentrations below 1 gram per cubic meter⁵ little if any difference is noted in the total ozone yield with increasing air flow. Therefore, if the line in Fig. 4 was extended it would strike an equilibrium, possibly at the 35 liter per minute mark.

The relation between the concentration of the ozone and the air velocity is shown in Fig. 5. From these two charts it will be noted that, as the air flow is increased the concentration declines, but the total amount of ozone produced is increased up to a point of equilibrium. A salesman of ozone equipment recently told the writer that the production of ozone was a straight line function of the air velocity through the generator, and that one of his units would supply the proper amount of ozone for 1000 cu. ft. of air per minute, but if the velocity through the unit was reduced to 500 cu. ft. per minute, the ozone would be proportionately reduced. Thus the same concentration would be supplied in both cases without altering the energy input. A definite concentration is certainly desired in ventilating work but the writer's work shows that it cannot be obtained so simply. This is not an exclusive discovery, however, for the two charts that have just been shown can be found, in one form or another, in practically every book that even pretends to treat ozone production with any degree of thoroughness. Schönbein was aware of this fact in 1850.

Oxides of Nitrogen

The oxides of nitrogen should not be present to any appreciable extent in ozone that is to be used for ventilation. The U. S. Bureau of Mines⁶ gives the following data:

Least amount to cause irritation: Nitric oxide, 62 parts per million parts of air, 101 parts per million parts of air, causing coughing; nitrogen peroxide the same.

Slight symptoms after several hours exposure: Nitric oxide 39 parts per million parts of air; nitrogen peroxide the same.

Dangerous in 30 to 60 min.: Nitric oxide 117-154 parts per million parts of air; nitrogen peroxide the same.

Impossible to breathe for several minutes: Nitric oxide 775 parts per million parts of air; nitrogen peroxide the same.

To say that it is impossible, with any type of ozonizer, to produce anything like the above quantities of oxide of nitrogen is by no means too broad a statement. In fact recent work on the determination of the oxides of nitrogen from ventilating ozonizers, conducted by the Bureau of Mines,⁷ demonstrated that none of the ozonizers tested produced oxides of nitrogen in more than negligible quantities. Our own ozonizers have never been found to produce oxides of nitrogen, calculated as nitrogen pentoxide, in a ratio greater than 1:100 of ozone, and to that extent only when operating under adverse conditions. Taking the highest concentration of ozone permissible in ventilating work, *vis.*: 0.1 parts per million parts of air, the concentration of nitrogen pentoxide would not exceed 0.001 parts per million parts of air.

The best conditions for the production of the oxides of nitrogen with the corona discharge are obtained when an enclosed volume of air is worked with. In the production of ozone it is not practical to employ enclosed volumes. High temperatures do not necessarily cause the formation of oxides of nitrogen, but as high temperatures are frequently noted, when the oxides of nitrogen are formed in the greatest quantities, this is often cited as a cause.

Some people adhere to the idea that the higher the voltage used, the greater the proportion of oxides of nitrogen formed. This is true above a definite limit for a given generator. The voltage used should be based on the air space between the electrodes and the dielectric constant of the dielectric material used. It should

be sufficiently high to yield an economical current density without causing too intense a discharge. If raised above that point, the discharge develops sparks and the production of the oxides of nitrogen is increased. The elevation of the voltage above a given point may increase the velocity of the electrons to such an extent as to afford them sufficient kinetic energy to activate nitrogen, which is not so good an electron trap as oxygen.

In our work it has been found that by ozonizing at elevated air pressures we can practically inhibit the formation of the oxides of nitrogen.⁴ When using elevated air pressures it is necessary to raise the voltage in order to obtain the same current density, developed at atmospheric pressure, since the dielectric properties of the gases are proportionate to the pressure. But the elevation of the voltage does

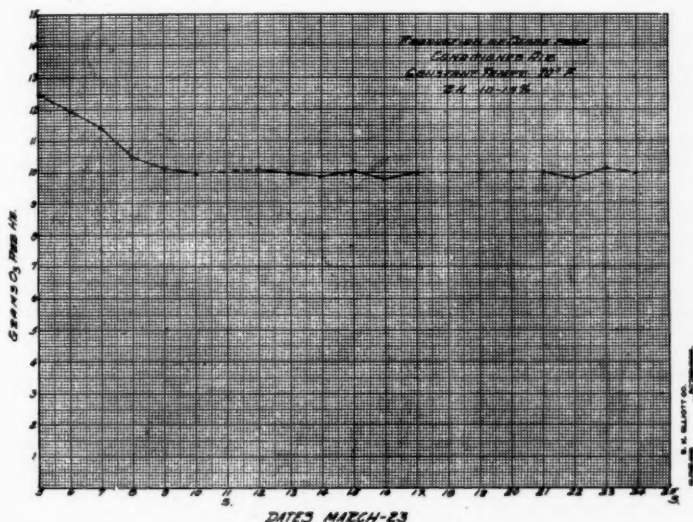


FIG. 3. CHART OF PRODUCTION OF OZONE FROM CONDITIONED AIR

not, in this case, change the character of the brush, since the current remains the same. It is seen, therefore, that the voltage used is no criterion of the oxides of nitrogen production. With clean, and reasonably dry air supplied to a properly designed generator there is no reason to fear the formation of the oxides of nitrogen.

Chemical Properties of Ozone

Ozone has a molecular weight of 48, a density of 1.658 as compared with air and possesses a characteristic odor not unlike freshly cut cucumbers or watermelons. Chemically, ozone is an exceedingly active oxidizing agent. It attacks all of the metals excepting gold and some of the platinum group and is extremely active in the oxidation of organic substances. It will oxidize phenol to CO_2 and water and attack the benzene ring. Ozone is not so active a bleach as is generally supposed, however, it possesses strong bleaching powers.

Ozone reacts two ways:

- a. $O_3 + M = MO + O_2$
- b. $O_3 + R = RO_2$

The latter way is exemplified in the oxidation of sulphur dioxide to sulphuric anhydride: $3SO_2 + O_3 = 3SO_3$

and we have also demonstrated that ozone reacts this way in the oxidation of organic matter during the purification of water. On the whole it may be said that the entire molecule of ozone is generally used in organic oxidations.

Mechanical, or Artificial Ventilation

Mechanical, or artificial ventilation has for its purpose the production in buildings, etc., or an atmosphere that is most conducive to health and comfort. There are several factors that bear on the quality of a ventilating system, the most important of them being *temperature and humidity control, air movement, odors and air change*. The elimination of dust is of economic significance and is advisable in the large industrial centers.

It is not intended that this shall be a treatise on the art of heating and ventilating, but an effort will be made to clearly set forth the various factors of importance, so that my treatment, of some of these factors, may be clear of misunderstanding. The object in view is the setting forth of the value of ozone as an adjunct to ventilation.

Each of the factors here considered represents a specific problem that requires specific treatment, and what may answer for one may be thoroughly inadequate for another. In a word, there can be applied no panacea to cure all of the ills that ventilating system is subjected to, and it should be well understood that ozone is of value as a *specific* for the treatment of specific conditions.

Temperature Control

An adult in good health radiates, in the course of an hour, 500 B.t.u. of heat, and a proportionately smaller amount is radiated by children. Temperature control, therefore, resolves itself into the removal of body heat at times, and again the supplying of sufficient heat, that when added to the heat radiated by the occupants of the room just produces a temperature that is best suited to the character of the work indulged in by the occupants of the room.

It is comparatively easy to maintain an unoccupied room at an even temperature, but when a room is subjected to variations in the number of occupants, or varying sources of leakage, etc., it is almost impossible to adequately control the temperature unless the thermostat be used.

An increase in temperature of the atmosphere of a room will produce a condition of pseudo-stuffiness, even when an adequate air supply is provided, and the relative humidity is maintained within the proper limits. This condition frequently attains in class rooms, auditoriums and offices where adequate temperature control is not provided and is productive, or lassitude and mental fatigue on the part of the occupants.

There is no substitute for temperature control, and, therefore, the thermostat is essential to all well-designed ventilating systems. Ozone has no effect on temperature, nor would it be advisable to use ozone as a corrective for conditions of inadequate temperature control. There is a definite relation between temperature, humidity and air movement, and the work of the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS is very illuminating

on this point, and the comfort line that they are now establishing should be carefully considered, and made use of in the design of a ventilating and heating system.

Humidity Control

As there is a specific temperature at which we can perform a certain task with the greatest degree of comfort, so is there a specific relative humidity at which we feel more comfortable and are induced to breathe more properly.

An excessive humidity is productive of depression and tends to render one less efficient in one's work. On the other hand an atmosphere that is too dry is equally as harmful. Humidity control, while not practiced to the extent of temperature control, is very important and has a specific bearing on the comfort of man.

The reason that humidity control is not practiced more than it is is due primarily to the fact that the humidity of a given climate is more apt to be constant, to a far greater degree, than the temperature, and, therefore, is not so much in the need of artificial control. The real reason perhaps lies in the prevalent use of thermometers. We are prone to judge climate by temperature alone. The hygroscope is little known to the laity, and we frequently suffer from the effects of humidity without knowing the cause of our discomfort.

The proper relation between humidity, temperature, and air movement, should be studied and properly phased if the maximum comfort is to be derived from humidity and temperature control. Ozone, per se has no effect on humidity, and should not be used as a curative for bad humidity conditions.

Air Movement

Air movement is very important, for without it much of the good that would ordinarily arise from an adequate air supply is lost. By air movement is meant the elimination of stagnant areas, or pockets, and the production of an air velocity not high enough to create a draft, but sufficient to break up the gaseous envelope that tends to surround one in stagnant air. This gaseous envelope is rich in effluvia from the body, and as most animal effluvia are very much heavier than air, these gases diffuse but slowly, and if not broken up by air movement, the occupants of the room will be forced to take in much foul air that could easily be avoided by providing good air movement. Without good air movement many of the ventilating measures are set at naught. Now ozone has a very decided effect on this evil, because it will oxidize, and thus destroy the gaseous products of the body and maintain the air free of odors and other contaminating substances. But I do not recommend ozone as a curative for bad air movement, simply because good air movement can be obtained by properly designing the ventilating system, without adding to its cost. Air movement is a matter of design rather than materials and if a ventilating system is adequate, it can be designed to give good air movement. To use ozone to correct bad air movement would be to use it to cover engineering faults, and such is *not* the true application of ozone.

Odors

Now comes a very important factor and one that has, so far, offered great difficulty in handling, especially where recirculation, or a curtailment of the air supply is desired during the winter months. The writer has frequently been asked, "What is an odor?" As this point is held in so much doubt, and is manifestly of such interest to ventilating engineers, I propose to devote some space to it here.

Primarily, the differences, and confusion of opinions, held concerning the nature of an odor apparently arise from a misunderstanding of the popular statement that it is yet undecided whether the sense of smell depends upon a chemical or a physical

process. It appears that some people misinterpret this reference to a physical process, and they are led to believe that the sense of smell is stimulated by energy emanations analogous to the radio active, or generally electrical in character. Those holding such views attempt to defend them by stating that a substance may give rise to prodigious quantities of odors without itself losing weight, and, therefore, odor must arise from an emanation rather than a volatilization of the substance.

Now it is well known among scientists that electrical and thermal stimuli do not give rise to olfactory sensations. Althaus states that electrical stimulation causes a sensation of the smell of phosphorus. Althaus' statement is not confirmed in the literature and may be based on a sensation of taste.

When the teeth have metallic fillings, and the body is charged with electricity, and electrolysis frequently occurs in the mouth and stimulates the taste bulbs. Taste is frequently associated with smell (*i. e.*) giving rise to a sensation of flavor and one sensation is commonly confused with the other. For instance, chloroform excites taste alone, but is frequently spoken of as the odor of chloroform, while garlic, asafoetida, and vanilla excite only smell, though it is customary to ascribing a distinct taste to these substances. So it is seen that one must be careful when referring to these closely allied senses and should always attempt to differentiate between them. It must also be borne in mind that, in the early history of the electrical science, the odor always perceptible around electrical machinery was called an *electric odor*, and this odor is frequently described in the literature as resembling phosphorus. Now it is well known that this odor is due to ozone and the oxides of nitrogen produced by static leakage and sparking.

It is generally conceded by physiologists that substance must be brought into direct contact with the rod cells of the olfactory membrane, in order that it may excite odor. As the free olfactory surface is always covered with a fluid secretion it is essential that the odoriferous substance be soluble in this fluid, in order that it may reach the rod cells. Such a consideration at once points to a chemical process. However, Venturi, Prevost and Liegeois have studied the well-known movement of odoriferous particles such as camphor, succinic acid, etc., when placed on the surface of water, and they suggest that a similar movement of such particles on the fluid of the olfactory membrane may cause an irritation that excited smell, and thus the stimulation would be mechanical or physical in character. This is the basis of the physical theory, and a very inadequate basis too, since with such an hypothesis it is indeed hard to account for the ability of the sense to differentiate between odors, and altogether is more apt than sound.

It has been pointed out that in order to stimulate the olfactory center, it is essential that the substance come into contact with the rod cells, also that these rod cells are protected by a fluid secretion. It is thought by some authorities that the odoriferous substance dissolves in the fluid and may exert a chemical effect that stimulates the sense of smell. This is the basis of the chemical theory, and since the sense of taste is considered to be stimulated through the solution of the sapid substances in the saliva, which in turn set up chemical action in the taste bulbs, it is reasonable to believe that the mechanism of smell is not very different, since the two senses are so closely allied.

We are now concerned with the nature of odoriferous particles, and how they reach the olfactory membrane. These particles are usually gaseous, or in a condition of vapor, and diffuse through the air, according to the well-known laws of gaseous diffusion. Non-volatile substances, or those having a very low vapor tension rarely excite smell, while the readily volatile substances such as iodine,

bromine, etc., always excite smell. All gases are not odoriferous, however, Sir William Ramsay has shown that the gases that possess a low molecular weight are odorless. Thus oxygen, hydrogen, nitrogen, etc., are odorless, while hydrogen sulphide, butane, etc., are distinctly odoriferous. This is well borne out in the paraffin series; methane (mol. wt. 16) has no odor; ethane (mol. wt. 30) has a slight odor, but it is not until butane (mol. wt. 58) is reached that a distinct sensation is noticed. The alcohols further confirm this for vapors. Methyl alcohol is odorless, ethyl alcohol has but a very slight odor, while butyl alcohol is distinctly odoriferous.

Now if the sense of smell is stimulated by direct contact between the substance and the rod cells of the olfactory membrane an odoriferous substance must be volatile, yet if the substance apparently does not lose weight, but at the same time

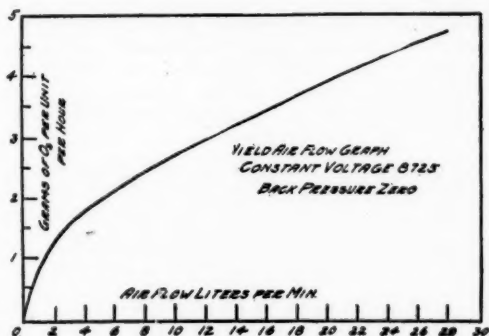


FIG. 4. YIELD AIR FLOW GRAPH

gives rise to prodigious quantities of odor, how is this apparent discrepancy accounted for?

To those unfamiliar with problems involving great effects from a very small amount of substance, it may seem astounding to say that a quantity of fluorescein, of the volume of 1 cu. in. is sufficient to color, to a distinct green, a body of water as large as our combined great lakes. This is true, however, and a consideration of it will assist in visualizing what is to follow.

When odors are perceptible the substance is present in the air either as a gas, a vapor, or in a state of exceedingly fine division. An idea of the fineness of odoriferous particles may be gained by conveying air containing an odor to the nostrils through a tube packed with cotton wool. It will be found that the odor is still discernible, and such a tube is capable of filtering from the air microorganisms of less than 1/100,000 of an inch in diameter.

A grain of musk will scent up an apartment for years without appreciable loss of weight. In fact, musk to an amount of two millionth of a milligram (1 milligram 0.0154 grains), or one part of hydrogen sulphide in 1,000,000 parts of air, may be perceived by the sense of smell. The smell of mercaptan has been detected when the dilution was 1:50,000,000,000, and it was calculated that the weight of mercaptan so detected was 1/400,000,000 of a milligram in 50 cc. of air (E. Fischer & Penzoldt).

From this it is seen that the sense of smell is exceedingly delicate, and with odors we are dealing with almost infinitely small quantities, therefore, the fact that a loss of weight is not detected can be ascribed only to our present crude methods of weighing. The foregoing considerations throw considerable light on odors in general, and indicate the necessity of specific measures for their elimination in a ventilating system, since so small an amount of substance can cause so much odor.

Before attempting the destruction of a substance one had best examine into its chemistry, in order to ascertain the reagent to which it is most susceptible. Since here the destruction must be carried out in the air breathed it is essential that the products of destruction be inoffensive and innocuous.

T. Graham has conclusively shown that odoriferous substances are susceptible to oxidation. The term oxidation has a much broader meaning since the advent of physical chemistry, and it is used today to generally indicate a gain of valence. For an instance, a reaction between chlorine and copper to form cuprous chloride may be spoken of as an oxidation reaction, yet no oxygen has entered into the reaction at all. Generally, however, the older meaning is indicated by the term, and oxidation is the act of adding oxygen. It is in its older sense that it is used here.

The addition of oxygen to a substance does not always mean rendering it innocuous. For example, a strong oxidation of allyl alcohol will yield the aldehyde acrolein, which is a very deadly gas, and in this case a much more objectionable substance is produced. This point is brought out simply to indicate the necessity of a careful consideration of any problem involving chemical reactions before attempting to apply reagents. To some, the terms oxidation and combustion are synonymous and interchangeable, but such is not the case, for the combustion of allyl alcohol is productive of acrolein. Recently the writer was approached for apparatus for the purpose of using ozone to oxidize a by-product in the form of an odoriferous gas, and the would-be purchasers of ozone equipment were inclined to resent our declining to make a sale. If this had been done, however, the use of ozone in this case would have made matters infinitely worse, for it would have produced a toxic substance from what is now simply annoying. These remarks apply only to definite industrial odors, and do not include animal effluvia, or putrefactive gases of animal or vegetable tissue. Broadly; the oxidation of these gases is more in the nature of a combustion, since the final products are generally CO_2 and water.

Putrefaction of animal and vegetable tissue is productive of such substances as the amino, aromatic, and fatty acids, indole, skatole, cresol, and also the alkaloid-like ptomaines, such as tetramethylene-diamine and pentamethylene-diamine, etc. The vapors given off from such substances cause the well-known odors of putrefaction. Of the substances mentioned those having an amino group contain nitrogen, while the remainder are hydrocarbons. The effect of ozone on the vapors of these substances has been studied and it is found that the effect is one of combustion: *i. e.*, the final products of the hydrocarbons being CO_2 and water and those containing nitrogen, nitrogen pentoxide in addition.

In the air surrounding places where putrefaction is going on, or the air from sewers, etc., while highly odoriferous, contain but traces of these substances and the odors are easily and completely destroyed with ozone without the slightest harmful result. Ozone has been successfully used in San Francisco to deodorize the air of a sewage pumping station, which, without ozone, became highly odoriferous when the occasion arose to clean out the pump screens. Recently a rat died in an in-

accessible wall of our office building, with the result that putrefaction at once began when air in several offices became unendurably odoriferous, but the application of ozone completely destroyed the odors, rendering them absolutely imperceptible.

Ventilation in general, however, hardly extends to the odors of sewage and putrefaction, but is concerned more with the odors of animal effluvia. These gases are mostly hydrocarbons, together with hydrogen sulphide. The effect of ozone on these hydrocarbon gases is one of combustion, producing as final products, CO_2 and water vapors. Hydrogen sulphide is oxidized to sulphuric anhydride and water by ozone, and ozone has been used extensively for eliminating H_2S from potable waters. The amount of H_2S thrown off by the human body is infinitesimally small so the amount of resulting sulphuric anhydride is negligible.

It has been said that the deodorizing effect of ozone is one of masking rather than destroying. The writer hardly thinks that such a statement can stand in the light of chemical inquiry, and the foregoing remarks quite conclusively establish its action as that of oxidation, which in the case means the same destruction as would be obtained by combustion. The only objection that can be raised is a questioning of the ability of ozone to react almost instantaneously at normal temperatures. This question the writer feels is amply answered by the work of numerous investigators into the purification of water with ozone. The organic matter in surface water is related to the types of organic substances treated here, and by actual chemical analysis it has been demonstrated that ozone will destroy this organic matter at the temperatures of surface waters. Further, the data that have been given here are based on actual experiment, and are not simply theorizing. In a recent conversation with an eminent ventilating engineer, he told me that he placed odors on the third position of importance, in a list of the vital factors effecting ventilation. I believe that we all realize the importance of this factor, and I believe that the ventilating engineer will welcome the use of ozone as a specific measure for the treatment of this evil. What the ventilating engineer has been waiting for is a means of quantitatively applying ozone, so that this treatment will be susceptible to visible control, rather than contingent upon the whims of nature, with the final results surrounded with mystery, being more psychological in its virtue than actual.

Physiological Effects of Ozone

In sufficient concentrations (say 15 p.p.m.) ozone is an exceedingly active respiratory irritant, and further attacks the mucous membranes of the eyes, nose and buccal cavities.

In low concentrations it is specifically remedial for chronic anemia, since small quantities of ozone increases the haemoglobin, or red corpuscle of the blood. It has been used in England with signal success for this purpose. Some exploiters of ozone have claimed a stimulating effect, and the purport of their remarks is that ozone is a great impartor of "pep," if I may use the word. My own belief is that stimulants are not good things, except in cases of acute illness. Although such a statement does not necessarily line me on the side of Mr. Volstead. Stimulation is always productive of reaction, and the reaction usually sets us back further than we were before the use of the stimulant.

It is well known that ozone is a natural constituent of naturally pure air. It is found in relatively large quantities near water falls, large bodies of water such as the Great Lakes, and the ocean, some excellent, and yet disproven claims, have

been advanced that ozone is a product of plant life, and it is known that nature provided the green leaf pigment to keep our air pure, and it is also formed by the action of sunlight on snow. The Swiss Alps are famous for ozone. Such places are well-known health resorts, and I do not believe that nature would resort to the administration of stimulants.

Ozone would be universally present in the air were it not for the congested cities that civilization has produced. The impurities of city air are so extensive as to use up naturally occurring ozone faster than it can be supplied, just the same way that our cities contaminate our water ways faster than nature can purify them. We never question the purification of water for this need is obvious to even the most uninformed. Then, is it not logical to purify our air by adding to it that substance which we demand faster than nature can supply? It is harder to ascertain the effect of impure air than it is to ascertain the effect of impure water. In this first place the impurities in air are so small that analytical methods have to be exceedingly refined; secondly, observations need to be made over a much longer period of time. Yet, simply because such contamination is elusive in detection, it does not follow that it is negligible in effect.

To return to the stimulating effect of ozone. The word stimulation has been used more for its aptness than its accuracy in describing the condition attained. Personally, I do not believe that ozone, per se, is stimulating, but I do believe that it is remedial, and has a specific value in the atmosphere that we breathe. I believe that the function of ozone is that of a scavenger, it keeps our air pure so that we will involuntarily breathe properly. Of all the automatic mechanism that modern genius and experiment have devised, there is nothing mechanical that any way near approaches the delicacy of the automatic control of the human body. The rapidity with which the pores of the skin respond to even minute changes in temperature has never been rivaled by the thermocouples or expanding gases or liquids.

As we pass from a warmer to a cooler atmosphere our skin automatically contracts so as to afford its maximum protection. A study of gland secretions is very illuminative of the extent of the automatic function of the body. This automatic control extends to our respirations. When the air contains even minute and practically imperceptible quantities of disagreeable odors, we involuntarily shorten our breathing, and though rate of respiration may increase under such conditions, the total volume of oxygen taken into our lungs is appreciably diminished. Our respiratory system has a definite, and important work to perform. All mental and physical exercise is accompanied by a destruction of tissue. As a result of this destroyed tissue impurities occur throughout the body, and if they are allowed to accumulate autointoxication will result. One of the functions of the blood is the elimination of this class of impurities from the system. Haemoglobin, which is the active red blood principal of the blood, serves as an oxygen carrier. In the lungs the haemoglobin comes into contact with the oxygen contained in the air that we breathe. This oxygen chemically unites with the haemoglobin forming oxyhaemoglobin and is carried throughout the body by the circulation of the blood. Through the capillaries the oxyhaemoglobin comes in contact with the destroyed tissue, and effects its combustion through ceding its oxygen content. The resulting gases of the combustion, such as water vapors, CO_2 , nitrous oxide, etc., in turn unite with the haemoglobin and are conveyed back to the lungs, where they are given up and expelled from the body in exhaled respired air. This cycle of events is continuous.

Now is it seen that the amount of oxygen that can be conveyed throughout our body is contingent upon the amount of aeration that our blood receives. If

foul air causes us to decrease the amount of air taken into our lungs it naturally follows that our blood will receive insufficient aeration, and, in consequence, impurities will remain in our system to cause autointoxication. Intoxication of this character is conducive to anemia, as the amount of haemoglobin will ultimately be decreased. Its immediate effects are headaches, and general lassitude that

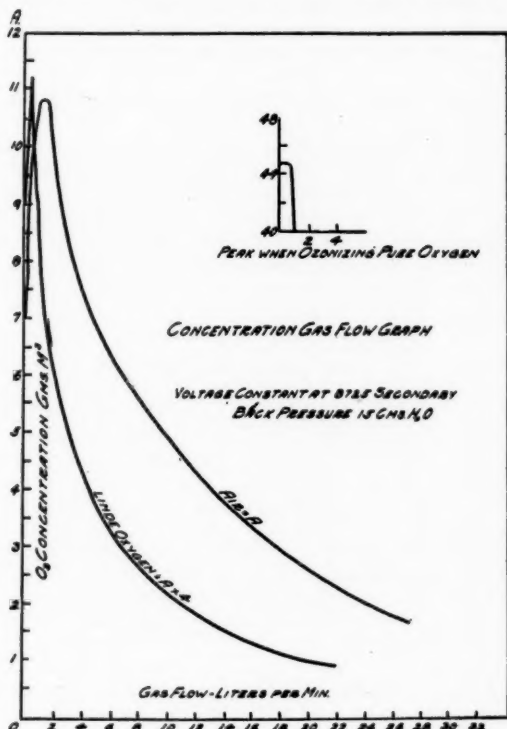


FIG. 5. CONCENTRATION GAS FLOW GRAPH

inhibits a free exercise of the mental faculties, and is further productive of physical fatigue.

Now, in well occupied rooms, such as the class rooms of schools, auditoriums, theatres, and large loft-like offices, it is indeed difficult to so design a ventilating system as to provide an unfailingly good air movement in every part of the room. The design of the system may not be at fault, and for an empty room may operate at a maximum efficiency. But I have observed that after a new building is occupied there is a tendency to accumulate furniture and fixtures, or relocate them. This is almost invariably done with no regard to the ventilating system, and it frequently occurs that unforeseen resistances are set up that entirely change the character of the air movement. Body effluvia are much heavier than the gases of the atmosphere, hence the diffusion of them is slow. Unless the air movement is

uniformly good an envelope of these gases is apt to form around a person, and thus he is forced to take in air that is more foul than the actual atmosphere of the room.

Minute traces of ozone in the air will serve to destroy these gases and break up this envelope. In this way the air is maintained in a pure condition, and in consequence, is at all times conducive to correct breathing. Correct breathing means well aerated blood, and well aerated blood will properly eliminate the impurities from the system and prevent autointoxication. Therefore, to sum up the case, ozone is a purifier and not a stimulant. Its use simply reestablishes a natural condition, and is not, therefore, a ready cure or stimulant in the strict therapeutic sense of the word.

A theory has been advanced that the presence of minute quantities of ozone in the air actually in the lungs is effective in reducing the partial pressure of the noxious gases in the lungs, and thus causing the blood to give up its impure gases more readily. This effect would be productive of two results. (a) A greater aeration; (b) An oxidation of the foul gases, *in situ*, so as to keep them from entering the atmosphere at all. The first result may account for the marked increase of oxy-haemoglobin that ozone produces in anemic persons.

Air Change and Recirculation

Air change involves the removal of definite quantities of air from room and replacing it with fresh air from the outside. This would be essential to avoid vitiation if a house was hermetically sealed. Vitiating, in so far as the carbon dioxide is concerned, is, under usual conditions, automatically taken care of through leakage, and this factor is better understood today than at former times. Air change is generally calculated to keep the air fresh and free from obnoxious gases and odors. Air changes and recirculation are so closely allied that a consideration of recirculation naturally involves a consideration of air change. Experiments that have been conducted on more or less extended scales all show that the recirculation of at least 75 per cent of the air used in ventilating a building can be accomplished without fear of increasing the CO₂ content to anything like a dangerous quantity. The only difficulty experienced in recirculation arises from the presence of odors in the recirculated air. When provisions are made for removing the odors, up to 75 per cent of the air can be recirculated without impairing its quality as tested by the Synthetic Air Chart. Ability to recirculate air is of great economic significance, as at least 50 per cent of the fuel required in heating can be saved. This is indeed a considerable item, as the present day condition of our fuel supply not only demands a conservation, but the prices asked necessitate economic measures wherever practical.

Ozone has practically demonstrated its value in recirculation as it accomplished the destruction of odors with great facility.

When ozone is used for recirculation in new installations, the saving in vent to will pay for the ozone equipment. The saving in fuel is an annual saving, and pays a very handsome dividend on the investment. The care and expense of a properly designed ozonizer are negligible.

Quantitative Application of Ozone

The writer has endeavored to point out the value of ozone in artificial ventilating. In recapitulation, it may be said the ozone is of great value in maintaining the air free from odors or organic gases that are foreign to naturally pure air. In doing

this ozone changes the air of the ventilating system into a condition as pure, in this respect, as the purest air of nature. That this is the desideratum of a ventilating system goes without saying, therefore, ozone supplies that element to the air that is essential to artificial ventilating, if a true simulation of nature is to be attained. Further, ozone has a great economic significance, in that it will permit of a recirculation of the major portion of air supplied the ventilating system and actually provide an improved air when so employed.

Ozone certainly behooves the careful investigation of all charged with the care

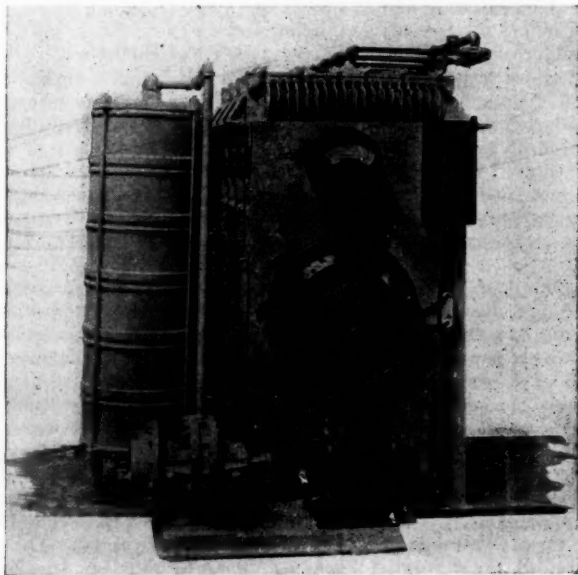


FIG. 6. MOTORED BLOWER WHICH SUPPLIES AIR TO TWO DEHYDRATORS AND COTTON FILTERS, THENCE TO THE OZONE GENERATORS

and administration of buildings, and this particularly applies to schools, and other public properties.

If unfailingly good results are to be obtained with the use of ozone, it should be applied quantitatively. In order to do this, it is necessary that the production of the ozone be at all times under definite control of the operator, and the character of this control needs be very simple, since the average ventilating ozonizer will be entrusted to unskilled hands.

There has recently been developed a means of ozone control, the operation of which involves no greater technique than the turning of a rheostat and reading an indicating instrument. There is no satisfactory physical method for the determination of ozone by direct methods. Otto's barograph serves very well in skilled hands, but is impractical for use in ventilation. The chemical methods of determining ozone, even the roughest field methods, are too time consuming to offer

practical means of commercial control. In my remarks on the production of ozone, I have referred to the external factors that bear on the output of an ozonizer. You will recall that dust and water vapors in the air to be ozonized are productive of erratic results. Temperature, over the normal indoor range, has but little effect. I have also pointed out that when the air to be ozonized has a relative humidity of 15 per cent or less, and is further free from dust, the output of ozone is a function of the energy input. Therefore, when these conditions are attained a measure of the energy is a measure of the ozone. It is obvious that we cannot reduce all of the air of the ventilating system to a R. H. of 15 per cent at any temperature, nor is the usual run of washed or filtered air sufficiently free from dust for ozonizing purposes. Therefore, it is obvious that the ordinary air of the ventilating system cannot be used for ozonizing, if quantitative results are to be obtained.

Our experience with ozone has been largely drawn from its application to the purification of water and the industrial chemical arts. In consequence, we have always worked with the view of producing ozone in definite quantities. This is not at all difficult to do with properly conditioned air, and a means of controlling the energy input. Instead, therefore, of placing units directly in the path of the air in the ducts, and attempting to produce ozone in concentrations just sufficient for ventilating purposes the ozone is produced in totally enclosed generators that utilize only a very small quantity of air for producing relatively high concentrations of ozone, and this is subsequently introduced into the fan inlet, in the proper quantities to produce the desired concentration in the circulated air. To give some idea of the small quantity of air required by the generators it is necessary to supply only 4 cu. ft. of air per minute in order to produce an ozone concentration of 0.1 part of ozone per million parts of air in 100,000 cu. ft. per minute of the circulated air. It is seen, therefore, that the question of properly conditioning the air for the ozone generators is not a difficult one, and its cost is negligible. Fig. 6, illustrates the ozone equipment necessary for producing a maximum ozone concentration of 0.1 part of ozone per million parts of air in 100,000 cu. ft. of air per minute. The motored blower shown in the illustration supplies the air to two dehydrators and cotton filters, and thence to the ozone generators. By thus properly conditioning the air to be ozonized, we can use a wattmeter calibrated in the terms of ozone for indicating the concentration produced at any given energy input. The energy input is controlled by a rheostat. Where the fans are operated at variable speeds we mark in red, on the scale of the ozone meter, the quantity of ozone to use with each of the fan speeds, and the operator has but to turn the rheostat until the meter indicates the proper quantity of ozone. When push button stations are used for definite fan speeds, we can phase our rheostat with the fan motor so that it is simply necessary to press the push button in order to operate both the fan and the ozonizer. The ozonizer will thus automatically cut on the proper quantity of ozone.

REFERENCES

1. Ozone, E. K. Rideal, D. Van Nostrand Co., New York, 1920.
2. Puschin & Kauchitshev, J. Russ. Phys. Chem. Soc., 46, 576, 1914.
3. Thesis of Dr. A. Starke.
4. F. E. Hartman, *Trans. American Electrochemical Society*, 1923.
5. Das Ozon, Fondrobert.
- 6,7. Bulletins, U. S. Bureau of Mines.

PERFORMANCE OF A WARM-AIR FURNACE WITH ANTHRACITE AND BITUMINOUS COAL

By A. P. KRATZ,¹ URBANA, ILL.

NON-MEMBER

THE tests reported in this paper were selected from the tests included in the general investigation of warm-air furnaces which is being conducted by the Engineering Experiment Station at the University of Illinois under a cooperative agreement with the *National Warm-Air Heating and Ventilating Association*. This investigation is being conducted under the general direction of A. C. Willard, professor of heating and ventilation, and head of the department of Mechanical Engineering.

Description of the Plant

The plant used for these tests is shown in Figs. 1 and 2, and consisted of a cast-iron circular-radiator type of furnace erected under a three story steel structure in the Mechanical Engineering Laboratory. This structure served as the working skeleton of a house, and carried the stacks and registers for the various floors. Such an arrangement permits the furnace to be operated under its own motive head, thus simulating conditions in a typical house installation in which recirculated air is used. All essential dimensions are shown in Figs. 1 and 2. A two-piece unslotted pot was used for all tests on anthracite coal, and a one-piece slotted pot for the tests on bituminous coal. For one of these tests the slots were sealed with fireclay.

Object of Tests

The tests were run with two objects in view:

1. To compare the performance of the same furnace operating with anthracite and bituminous coal.
2. To determine the effect of the use of a slotted firepot on the operation with bituminous coal.

Method of Testing

All temperature measurements were made by means of thermocouples and a potentiometer. The thermocouples were calibrated in place. The volume of air

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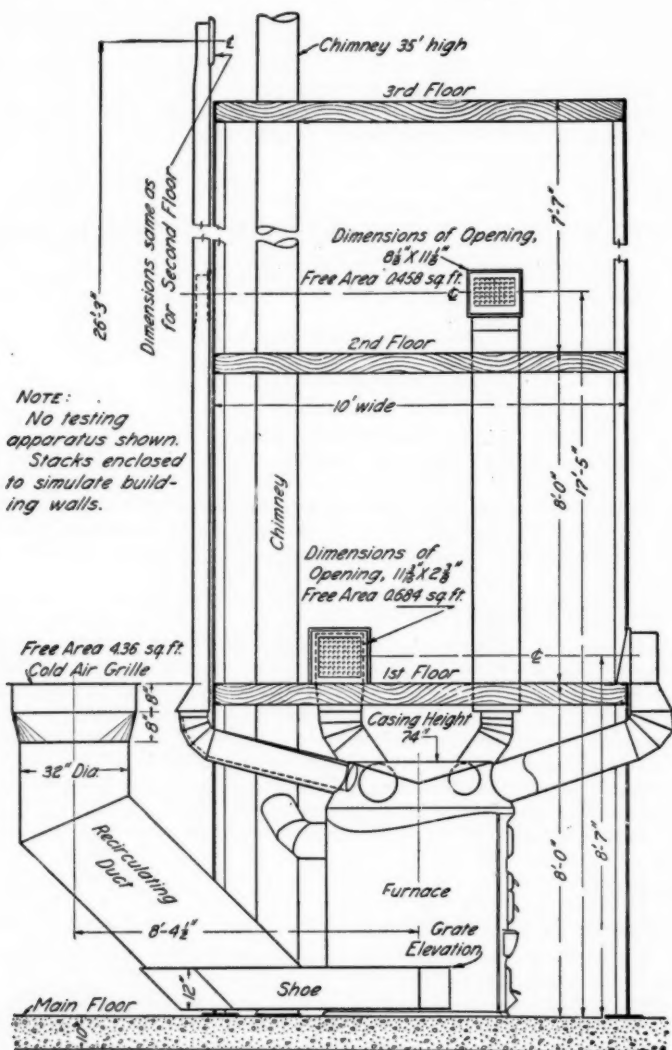


FIG. 1. SECTIONAL ELEVATION OF THE PIPED FURNACE TEST PLANT

flowing was measured with an anemometer located at the center of the recirculating duct as shown in Fig. 2. The anemometer was calibrated by placing it in a similar

duct connected to the suction side of a fan by a 10-in. round duct, and comparing its readings with the readings taken with Pitot tube in the 10-in. duct. The drafts were measured by means of a recording draft gage connected differentially between the ashpit and the smoke outlet.

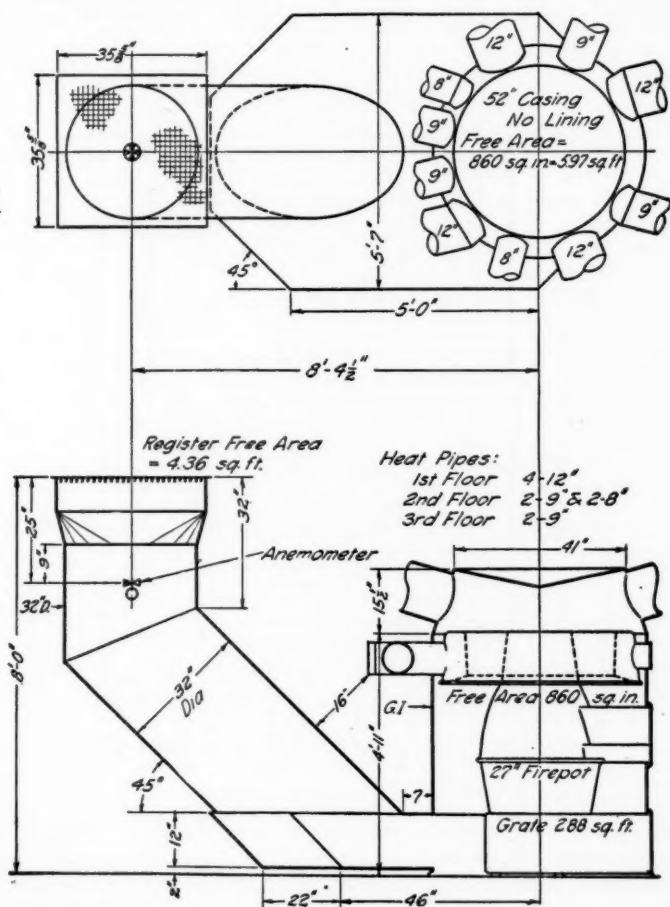


FIG. 2. PLAN AND ELEVATION OF THE FURNACE AND RECIRCULATING DUCT

For the tests on anthracite coal, the fire was started on clean grates using a charge of wood equal to 10 per cent of the coal charge required. A full charge consisting of sufficient coal to fill the firepot to the level of the bottom of the feed neck was used. The wood was allowed to burn for 10 min., and then the coal charge was

divided into three or four lots and fired so that the total charge was in the furnace by the end of the first hour of the test. The fire received no further attention until the close of the test. When about 80 per cent of the fuel had been burned the fire was quenched. The residue was dried and weighed and then reduced to terms of equivalent coal. The latter was subtracted from the weight of the original coal charge in order to obtain the weight of coal burned. The dampers were auto-

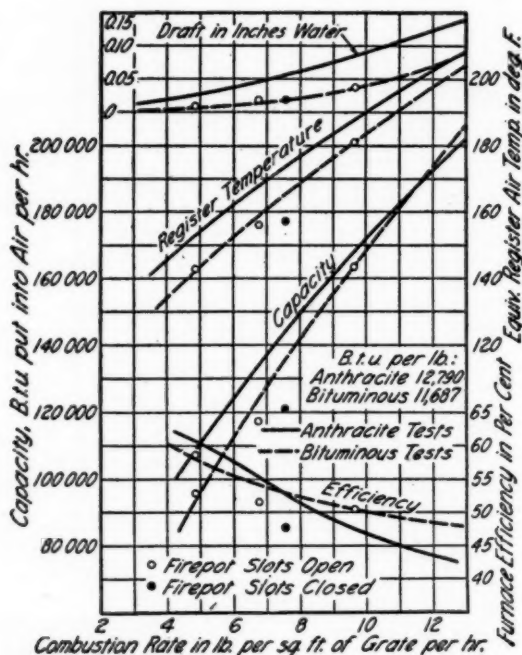


FIG. 3. PERFORMANCE CURVES OF FURNACE FIRED WITH TWO TYPES OF COAL

matically controlled so that a constant temperature of the air at the bonnet was maintained throughout the test.

In the case of bituminous coal, it was not feasible to fire the entire charge within the first hour of the test. Small charges of fuel were fired at approximately regular intervals, depending on the rate of combustion, and the fire was leveled between firings when the flue gas analyses indicated that holes had developed in the fuel bed causing an increase in the amount of excess air. Very little auxiliary air was admitted through the damper in the fire door, and the surface of the fuel bed was never allowed to burn appreciably lower than the tops of the slots in the firepot. The furnace was fired over a preliminary period of several hours before the start of a test, and conditions were maintained the same, as

those required for the test. At the start of the test, the condition of the fuel bed was noted and the ashpit cleaned, and the test was closed with the fuel in as nearly the same condition as possible. A test period of approximately 36 hours was required in order to reduce the error in estimating the conditions of the fuel bed within a negligible percentage of the total fuel burned. At the close of the test the ash and refuse in the ashpit were removed, weighed and analyzed. From the weight and chemical analyses, the coal equivalent was calculated for the ash and refuse, and this was then subtracted from the weight of coal fired.

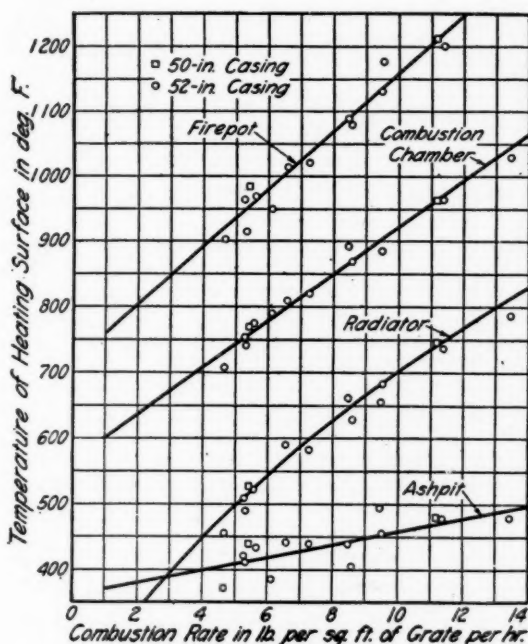


FIG. 4. TEMPERATURE OF THE HEATING SURFACES FOR THE FURNACE WHEN BURNING ANTHRACITE COAL

The chemical analyses of the coal, and of the ash and refuse are presented in Table 1.

Results and Conclusions

Performance curves for the furnace fired with two types of coal are shown in Fig. 3. From these curves it appears that:

1. Within practical combustion rates the anthracite coal gave higher efficiency and capacity (based on rise in the temperature of the air from inlet to bonnet) than the bituminous coal for the same combustion rate.

At combustion rates that are excessive for warm-air furnace practice, however, the reverse was true.

2. In the case of the bituminous coal, the efficiency was more nearly constant over the whole range of combustion rates than it was for the anthracite coal.

3. At a given combustion rate, more draft between the ashpit and

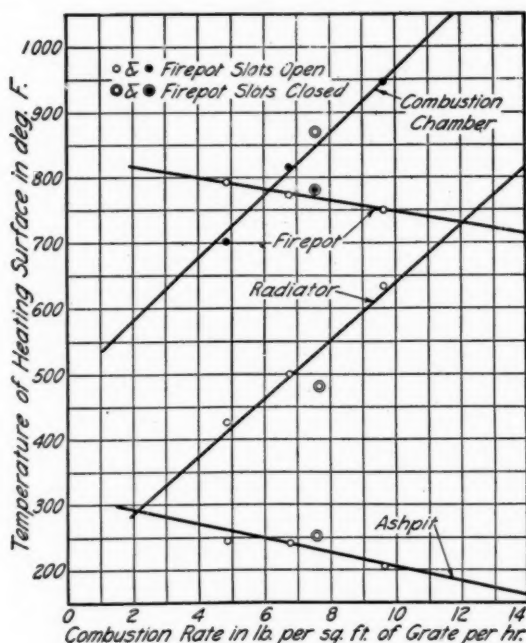


FIG. 5. TEMPERATURE OF THE HEATING SURFACES FOR THE FURNACE WHEN BURNING BITUMINOUS COAL

smoke outlet was required to operate the furnace on anthracite coal than on bituminous coal.

4. With bituminous coal, as fired under the conditions of the tests, the slotted firepot gave about 9 per cent greater efficiency and capacity than the firepot with the slots sealed.

The temperature of the heating surfaces of the furnace when fired with anthracite and bituminous coals, respectively are shown in Figs. 4 and 5. These curves indicate that the firepot temperatures increase with an increase in combustion rate for the anthracite coal, while the reverse is true for the bituminous coal. This decrease in temperature indicates that the higher drafts required to produce the higher combustion rates cause an increase in the amount of air drawn through the slots, and result in cooling the firepot. The temperature of the firepot was uni-

TABLE 1. CHEMICAL ANALYSES OF COALS AND ASH AND REFUSE

Proximate Analysis of Coal as Fired		Anthracite	Bituminous		
Fixed carbon, per cent.....		78.98	45.01		
Volatile matter, per cent.....		6.19	36.79		
Moisture, per cent.....		1.44	6.40		
Ash, per cent.....		13.39	11.80		
Calorific value by oxygen calorimeter, B.t.u. per lb.....		12,790	11,687		
Ultimate Analysis of Coal as Fired					
Carbon (C), per cent.....		79.50	64.69		
Hydrogen (H), per cent.....		2.43	4.27		
Oxygen (O), per cent.....		1.68	7.00		
Nitrogen (N), per cent.....		0.75	1.57		
Sulphur (S), per cent.....		0.81	4.27		
Moisture at 105° C., per cent.....		1.44	6.40		
Ash, per cent.....		13.39	11.80		
Analysis of Dry Ash and Refuse for Tests on Bituminous Coal		Test 98	Test 99	Test 100	Test 101
Fixed carbon, per cent.....		5.47	6.07	7.58	5.71
Earthy matter, per cent.....		94.53	93.93	92.42	94.29
Calorific value, B.t.u. per lb.....		799	886	1106	934

NOTE.—Anthracite coal all stove size. Bituminous coal run of mine, lumps broken to stove size.

formly lower for the bituminous coal than for the anthracite coal. The firepot temperature was also materially lower for the slotted firepot than for the one with the slots sealed. The fact that the temperature of the combustion dome was greater for the bituminous coal, indicates that more combustion took place above the bituminous coal fuel bed than above the anthracite bed. On the other hand, the radiator temperatures seem to indicate that the combustion in the case of the bituminous coal was more retarded in the radiator, and that probably the loss due to combustible in the flue gas was greater than it was for the anthracite coal.

Accumulations of soot in the radiator in the bituminous coal tests amounted to about $1\frac{1}{2}$ lb. at low combustion rates and were negligible at high rates. At low rates, the soot collected on the inner surface of the radiator to a depth of about $\frac{3}{8}$ in. while at high rates the surface remained practically free from soot.

SELECTING WALL STACKS SCIENTIFICALLY FOR GRAVITY WARM AIR HEATING SYSTEMS

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NON-MEMBER

Selecting Wall Stacks

IN a well-designed gravity warm-air heating system careful consideration must be given to the selection of proper wall stack sizes as well as to the selection of horizontal leader or basement pipe sizes. It is more essential that care be used in the selection of wall stacks than of leader pipes because leader pipes may be made greater in area than the stacks which they supply, but for obvious reasons the stacks cannot be increased in size at will, and must therefore be the controlling element in the planning of a correct heating system.

It has been shown by tests² that there is little to be gained by making the ratio of stack area to leader area exceed 0.75. On the other hand emphasis has been placed by heating engineers upon leader area rather than stack area. A more logical method of design would be based on the selection of the controlling element in the system of the stacks.

There are ample data for the design of stack sizes, based on values determined experimentally in the research of the *National Warm Air Heating and Ventilating Association* at the University of Illinois. In one phase of this research the heating characteristics of a number of stack sizes were experimentally determined.³

From these experimental results there have been plotted, by the interpolation between experimental curves, the curves of Fig. 1, showing the heating effect obtainable at the registers for a variety of stack areas and register air temperatures. The curves are plotted against register air temperature because that temperature is a fundamental starting point in the design of the system. It bears a definite relation to the velocity of flow in the stacks and pipes, and to the efficiency and capacity of the heater, and it is the physical factor which distinguishes between a well designed warm-air system and a poorly designed hot-air system. Moreover, it is a convenient basis upon which to design.

Fig. 1 may be easily and effectively applied in the selection of wall stack sizes.

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² Bul. 141, Engineering Experiment Station, University of Illinois, *JOUR. A.S.H.&V.E.*, July 1923, p. 407.

³ *JOUR. A.S.H.&V.E.*, July 1923, p. 407; also Bul. 141, Engineering Experiment Station University of Illinois.

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The calculation of the heat requirement of rooms is a matter well known and used by the heating profession. The register temperature is predetermined, and from the curves of Fig. 1 the cross-sectional area of the stack which will convey the required amount of heated air at the proper register air temperature may be selected. Conversely, if the size of stack is predetermined, the heating effect available may be read from the curve, or again, if the size of the stack is fixed and the heating requirement known, the necessary register air temperature may be read from the curve.

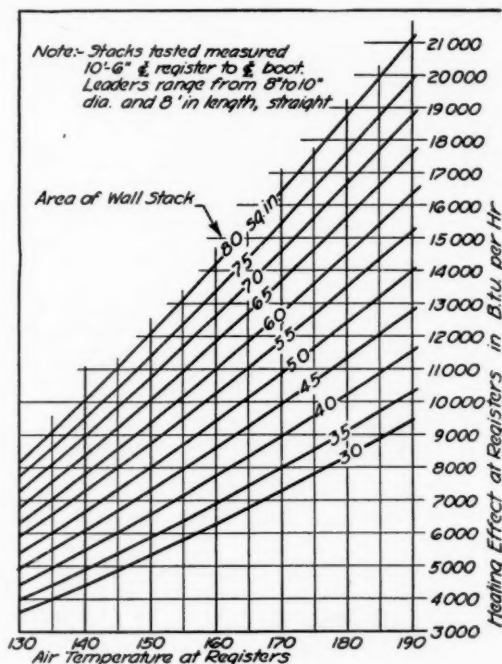


FIG. 1. RELATION BETWEEN REGISTER AIR TEMPERATURE, STACK AREA AND HEATING EFFECT AT REGISTER

For example, a room having a calculated heating requirement of 10,000 B.t.u. per hour may be heated, at 170 deg. fahr. at the register, by a stack 45 sq. in. in area. If a register air temperature of 150 deg. fahr. is used, the stack must be 65 sq. in. in area, or if the stack may not exceed 35 sq. in. in area, the register temperature must be 188 deg. fahr. Thus a given heating effect may be delivered in large pipe at a low register temperature, and in a small pipe at a high register temperature.

Selecting Register Air Temperature

It becomes important to know, therefore, the considerations affecting the selection of register air temperature. Apparently any register air temperature between the limits of 120 deg. and 220 deg. which have been the limits found in tests, could

be used as a basis for designing a heating system. This is theoretically true, but a simple calculation will show that as the temperature is decreased the volume of heated air necessary to make up the heating requirement is considerably increased and simultaneously the velocity of flow is decreased by the reduction in motive head in the system caused by the reduced temperature differential between warm and cold sides of the system. The pipe size must be proportionally increased for both factors. (The curves of Fig. 1 account for both the volume and velocity changes, being experimentally determined.) It is evident, therefore, that the register temperature selected as a basis for design must not be so low as to require

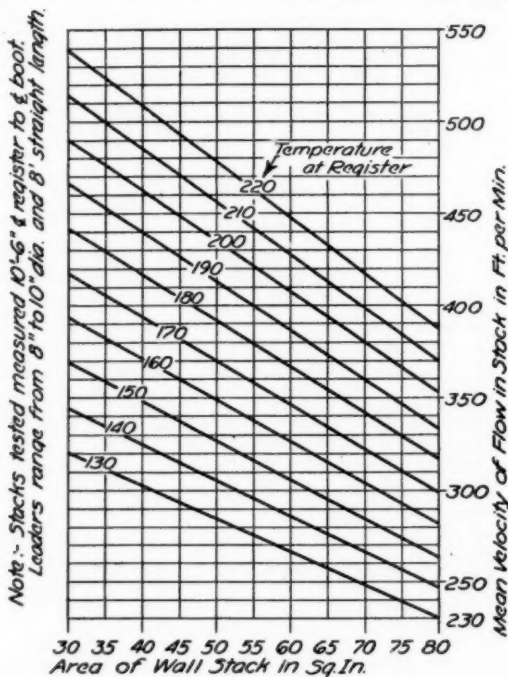


FIG. 2. RELATION BETWEEN STACK AREA, MEAN VELOCITY OF FLOW IN STACK, AND REGISTER AIR TEMPERATURE

very large and impractical pipe sizes nor so high as to give high discharge velocity with its accompanying high ceiling temperatures. The temperature at the register must be a reasonable one.

Velocity in Stacks

Fig. 2 has been prepared to show the velocities of flow actually obtaining in the stacks of gravity warm air furnace installations. The chart shows the relation between register air temperature stack cross-sectional area and mean velocity of

flow in the stacks. The velocities of flow were calculated from the same experimental data as were the values in Fig. 1.

With Fig. 2 the actual amount of the changes in velocity occurring in stacks due to changes in temperature may be determined. In the foregoing example, a register air temperature of 170 deg. fahr. and a stack 45 sq. in. in area were mentioned. Fig. 2 shows the velocity to be 381 ft. per minute. If, as before the temperature was changed to 150 deg., the stack area would become 65 sq. in. and Fig. 2 shows the new velocity in the stack to be 295 ft. per minute. Thus it is evident that reducing the register air temperature 20 deg. would bring about a reduction in velocity of 86 ft. per minute, or 23 per cent.

A good designer may use standardized formulae for the design and calculation of wall stack and pipe size, but as a check in the case of special rooms he will calculate the velocity of flow in the pipes to be sure that the amount of air required to make good the heat losses from the room will not necessitate velocities of flow which are not obtainable or practical. Fig. 2, the velocity chart, will be especially useful for this purpose because it shows the velocities obtainable with given stack areas and register air temperatures.

Calculating Velocities in Stack

The velocity of air flow in the stack required to make good the heat losses from the room may be calculated from the following:

$$V = \frac{10H}{A \times d_r \times (t_r - 70)} = \text{velocity in ft. per min.}$$

in which H = heating requirement of room, in B.t.u. per hr.
 A = area of stack in square inches.
 d_r = density of the warm air.
 $(t_r - 70)$ = difference in temperature between register and normal room temperature.

This expression gives values of velocity of flow at the register slightly lower than the mean velocity of flow in the stack, the difference being on the side of safety.

The application is as follows: If, as before, the heat requirement H is 10,000 B.t.u. per hour, and register temperature 170 deg. fahr. and the stack area 35 sq. in., the velocity is

$$\frac{10 \times 10,000}{35 \times 0.0610 \times 100} = 470 \text{ ft. per min.}$$

Reference to Fig. 2 will show that 405 is the velocity corresponding to 170 deg. for this size stack and a change must be made as the velocity required exceeds the velocity obtainable. If the stack area is fixed the temperature must be increased. If 180 deg. is assumed, the velocity calculated will be 430 which is in agreement with the value shown in Fig. 2 at 180 deg. fahr. On the other hand, if the register air temperature is fixed at 170 deg. fahr., the stack size should be increased. If 45 sq. in. is assumed, the velocity calculated will be 365 and Fig. 2 shows that the stack is capable of developing a velocity of 380 ft. per minute.

It is important to notice in Fig. 2 that higher velocities of flow may be obtained in small stacks at low temperatures than can be obtained in large stacks at much higher temperatures.

The careful designer of gravity warm air heating systems will find Fig. 1 of exceedingly great value in the direct determination of stack sizes, and Fig. 2 will be useful to him in the determination of the stack velocities. Both size and velocity are shown to be fundamentally dependent upon *register air temperature* which must, therefore, be the basis of design in gravity warm air heating conductors.

PRACTICAL APPLICATIONS OF THE HEAT FLOW METER

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MEMBER

THE object of this paper is to give the results of the application of the Heat Flow Meter described in the January, 1924, JOURNAL OF THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS for determining the thermal constants of certain building structures. As outlined in the previous paper, the readings of the meter at any moment give the rate of heat flow through it at that instance. Placing one on a surface therefore gives the rate of flow into it, and by taking such records over a period of time the total and average flows are known. If in addition the necessary temperature measurements are made, the thermal conductivity can be determined, as well as the surface coefficients, provided all the heat measured as entering the wall has also passed out, and provided the wall has the same temperature distribution through it at the start and finish. As far as temperatures could be measured at various depths of the wall such limitations would not necessarily be required, but such a procedure is not usually practicable.

Under the varying in and outside temperature conditions that exist naturally, it is therefore necessary to make the observations continuously over a sufficiently long time that any change in heat content of the wall itself may be negligible, and also preferable to cover a cycle of temperature changes such that they are nearly the same at the beginning and end of the test. In addition a rough correction can be made for any difference in the heat content of the wall. Heavy walls have a large heat storage capacity. For instance a 24 in. concrete wall stores up in it for 1 deg. Fahr. rise as much heat as passes through it in six hours under winter conditions.

The time required for a test depends mainly on the weather. In a district where it is uniformly cold and the wall reaches approximately constant conditions it can be shortened, but where it is variable and the wall is heavy, it must be run continuously for a considerable time. It also depends on the constancy of the inside temperature if the meter be on that side of the wall, since any small change at once alters the rate of heat flow, whereas an outside change will not be reflected through for some time and will be ironed out by its heat capacity.

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It is further evident that in the averaging out of heat flows and temperature for a period it is easier to get a true average when the values are large, and that though tests could be carried out during mild weather, yet the test must then be longer. The short period of the comparatively mild winter just passed was therefore used to test a variety of different constructions rather than continuously following up one as would be the better plan. All those tested were parts of the buildings of the U. S. Bureau of Mines at Pittsburgh, which offered a fair variety. These buildings were erected about 1916, and therefore are sufficiently old to have reached a permanent condition.

The method of making the tests was practically the same for all, both in the set up and temperature observations. The meter plate was on the hot side of the wall and puttied around the edges to prevent any air circulation between the surfaces. The temperatures taken are shown in Fig. 1, and for each location there were from three to five thermocouples in parallel so as to get an average. In addition each of the individual couples could be read if desired so as to note the variations for any one set of readings. Also the reading between two sets of thermo-junctions,

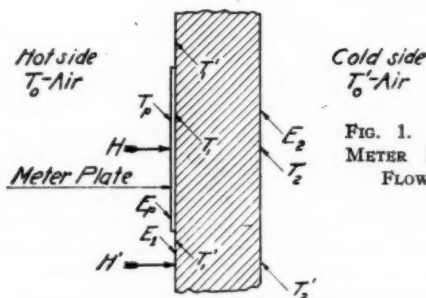


FIG. 1. ARRANGEMENT OF METER PLATE FOR HEAT FLOW OBSERVATIONS

such as T_1 and T_p , could be obtained directly. The thermo-junctions for T_1' were distributed around the plate, giving the average natural wall temperature. The air temperatures were taken five inches from the wall faces.

The observed heat flow through the plate is indicated by H . The corresponding flow H' which occurs through the natural wall will be slightly larger than this due to the thermal resistance of the plate itself. If the observed values are averaged over the whole test time it is given by:

$$H' = \frac{T_1' - T_o'}{T_1 - T_o} H$$

since the surface coefficient E will have the same value all over the outer wall. From this the surface coefficients E_1 and E_p can be obtained and, within certain limitations, can be used to estimate instantaneous values for H' corresponding to similar values of H .

The longest period of continuous observation was eleven 24 hr. days. The weather conditions in Pittsburgh are too variable to be ideal for such work so that advantage had to be taken of cold spells as they occurred. Attempts were made to keep the inside temperatures constant, but as the rooms were in their ordinary use, and the radiators large, this was not very successful. Readings for the heat flow were taken every half hour, and every hour for the temperatures.

The structures tested were:

A. Concrete foundation wall 26.2 in. thick including $\frac{3}{4}$ in. cement plaster. North exposure, meter plate located 12 ft. from outside ground level. A sill of a large window was 10 ft. directly above the plate. The specification for the concrete was 1-2-4, the aggregate to pass $1\frac{1}{2}$ in. mesh. The outer face was $1\frac{1}{2}$ in. thick surfacing coat, with rough finish. The inner surface of the concrete was asphalted before the plaster finish was applied.

B. Brick wall $22\frac{13}{16}$ in. thick, including $\frac{3}{4}$ in. cement plaster finish, consisted of one layer of hard burned, repressed Kittanning face brick, No. 3 shade grey, and the rest common red. No samples of those actually used were obtainable. The face brick had one row of headers every ten rows, which was included in the test. The bonding of the red brick was not known.

C. Hollow brick wall shown in Fig. 2. Bricks same as B. West exposure. Meter plate 10 ft. above ground, and 9 ft. above floor. The plate was 4 ft. above bottom of hollow space which the drawings showed to extend up for 30 ft. without closing off horizontal courses.

D. Brick wall of garage $8\frac{3}{4}$ in. actual thickness, painted inside, but not plastered. One layer of hard, burned, repressed Kittanning face brick, No. 3, grey shade, and one layer of common red. Face brick 5.78 lb. per brick or 137 lb. per cu. ft. Built without headers.

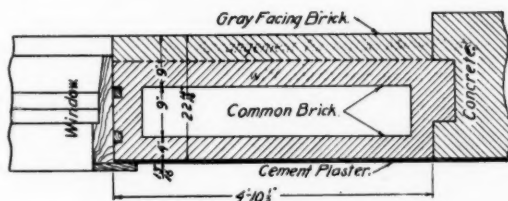


FIG. 2. SECTION OF HOLLOW WALL C

E. Same as D, except plate 7 ft. 6 in. above floor and 8 ft. 6 in. above ground and located near a row of heating pipes.

F. Plaster ceiling with metal lath and wire netting support. Plaster about $\frac{3}{4}$ in. thick, with rough, lumpy upper side. The location of the meter plate for this as well as for G and H is shown in Fig. 3. The attic was unused and had no definite openings to the rooms below beyond those for the light fixtures. The supporting metal lath was $\frac{3}{4}$ in. square stock set at 12 in. centers.

G. Partition wall composed of 4 in. thick by 2 ft. 6 in. by 12 in. pyrobar blocks, and $\frac{3}{4}$ in. plaster. The location of the meter plate is shown in Fig. 3. The rear side was unfinished and open to the loft. The heat flow through it was low.

H. Roof composed of $2\frac{1}{4}$ in. tongue and grooved boards covered with $\frac{3}{16}$ in. tar roofing, and $\frac{3}{16}$ in. thick slate. Details are shown in Fig. 3 which also shows the portion of the building for tests of sections F, G and H. The air gaps and contacts will have an appreciable insulating value in a construction of this type and would here be approximately equivalent to a $\frac{1}{8}$ in. air gap resistance.

A summary of the results obtained are given in Table 1 and express the average values from each test. As each test was run so as to cover as nearly as possible a cycle of temperature changes, the wall temperatures at the end were for all tests very closely the same as those at the start. Where there was any change the small correction necessary was applied. The conductivity coefficients are fully defined and in addition their nomenclature as suggested for general use is given. (E. F. Mueller, *Refrigerating Engineering*, Oct. 1923, p. 133.)

The discussion will first be confined to the conductivity coefficients found, and the surface and overall coefficients will be treated later.

The conductivity of 11.3 for concrete is higher than the values obtained by various investigators using small specially built samples. These have of necessity been thin, and consequently have not permitted a large aggregate to be used. When

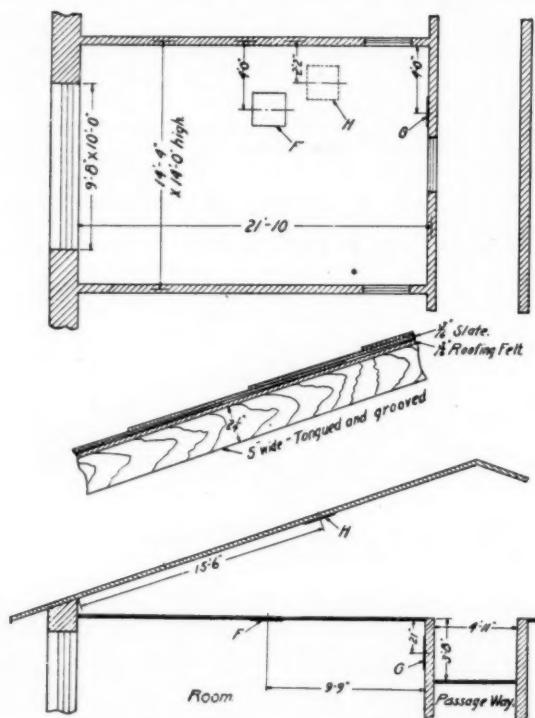


FIG. 3. TEST OF CEILING, WALL AND ROOF F, G AND H

thin and tested under laboratory conditions they would become well dried. Also it is to be expected that a large wall will have a higher density than can be obtained in a small sample. The value most commonly used is probably 7 though test results have shown from 5 to 9. Recent tests by Kreuger and Eriksson of Sweden on $6\frac{1}{2}$ ft. square samples called concrete, but made of 1 cement to 5 sand in order to get a homogeneous material gave the following:

Thickness	k
4 in.	9.4
8 in.	9.2
16 in.	9.4 to 9.6

As to whether a full sized but thinner wall would show the same high conductivity has yet to be developed.

A comparison of the brick walls is interesting. There is no clear reason why the thick wall should show a higher conductivity than the thinner ones, the face brick is the same to all records and appearance, although the samples of those actually

TABLE 1. AVERAGE VALUES FROM EACH TEST

DESIGNATION	A	B	C	D	E	F	G	H
CONSTRUCTION	Concrete 20.2" thick	Solid Brick 22 1/2" thick	Hollow Brick 22 1/2" thick 9" air space	Solid Brick 8 1/2" thick	Solid Brick 8 1/2" thick	Plaster ceiling metal support	Partition wall of Pyrobar	Slate roof with 2 1/2" wood
(See details in text.)	North	North	West	North	North	West
Exposure.....	44	46	45.4	46.2	46.8	72	68.2	44
Av. temp. of material, deg. fahr.....	41.4	44.4	40.3	40.4	40	16	13.9	22.8
Av. temp. diff. air to air, deg. fahr.....	25.5	33.6	26.45	22.8	23.6	...	5.9	19
Av. B.t.u. flow through plate and wall.....	10.68	8.66	8.7	11.74	12.94	8.28	3.13	6.13
Av. B.t.u. flow through uncovered wall.....	11.03	8.8	8.83	13.2	13.45	9.61	3.2	6.81
(Commercial) conductivity = k (a).....	11.35	6.14	5.21	5.03	...	2.4
Conductance, i. e., conductivity face to face = C (b).....	0.435	0.261	0.334	0.596	0.575	...	0.543	0.35 0.33 (c)
Transmittance, i. e., conductivity air to air = U (b).....	0.267	0.198	0.219 0.21 (d)	0.327	0.333	0.6	0.23	0.254 (d)
Surface Trans- mission = E (b)	Inner hot side 1.03 Outer cold side 2.17	1.13 2.87	0.91 2.1 1.46 (d)	1.11 2.06	1.27 2.2	1.4 ...	0.75 0.96	1.64 1.3 (d)

(a) B.t.u. per sq. ft., per hour, per inch, per deg. fahr.

(b) B.t.u. per sq. ft., per hour, per deg. fahr.

(c) When roof is covered with snow.

(d) The outside air temperature is taken as the average of that in the north shade. These values thus give a kind of practical coefficient which includes the type of exposure.

used were only available for the thin walls, tests *D* and *E*. The actual make of common brick was not known. The most probable cause would be the fact that *D* and *E* had no brick bonds and would thus have the advantage of the relatively higher insulation due to the mortar or a 3/4 in. air gap between the bricks. On the other hand it might have been expected that these would have been more subject to infiltration, although the wind was not consistently high on any of the tests and generally more western, whereas all of these walls had northern exposure.

All tests on brickwork have shown variable values even in laboratory tests, and it must be expected that it is more likely to be thus in walls as built. As to whether moisture plays any part in increasing the conductivity of thick walls is yet to be proved, although there would be that tendency.

The hollow brick wall, *C*, has the same total thickness as the solid one, *B*, but has a conductivity face to face of 0.328 against 0.267 for the solid. The apparent conductivity of the hollow wall can be estimated from its component parts provided these are known, but not much reliance could be placed on such estimation in close comparisons as neither the conductivity of the brick or air space can be predicted close enough for the actual conditions. Taking $k = 6.14$ for the brick, and 0.7 for the air space (Kent, p. 629) the total factor figures as 0.294, which is larger than that of the solid wall. Thus on simple considerations there is no extra insulation obtained by the use of the 9 in. air space. It would seem probable how-

ever that an air space like this has in it less insulating value than that equivalent to the 0.7 factor assumed especially near the bottom of a large height. It is ideal for encouraging infiltration and the cold air would collect at the bottom.

Due to the roughness of the upper side of the ceiling plates, F , no attempt was made to measure that surface temperature, and thus the conductivity of the plaster itself was not determined. The over-all conductivity of 0.6 is however the practical figure which is required.

Two values are given for the roof. The period when snow remained on it did not last long, and therefore that value is not as assured as could be desired. When not covered with snow the wind will get below the slates and will partially destroy the insulating value of the air space between them and the roofing, so that the conductivity would be increased. The conductivity found can be compared with that estimated for the component parts by taking the probable values of k for the wood 0.9, for the roofing felt 0.7, for the slate 13, and for the $\frac{1}{4}$ in. air gap, a conductance factor from surface to surface of 1.85. These values can be found in Kent (p. 629 and 630). The resulting conductance from surface to surface figures as 0.27. Considering the uncertainty of the air gap, and the shrinkage of the boards it is reasonable that it should be lower than actual values.

The heat flow for the partition wall H was low, and it is rare that one would exist with plaster on one side only.

The surface transmission coefficients are interesting. The values obtained in laboratory tests have usually been for similar conditions, and therefore do not show much variation. It is the custom to express it as the B.t.u. per square foot per hour divided by the difference between the surface and air temperatures, which at once raises the question as to where the latter is to be taken. The values given are all for about 5 in. from the surface, except for the roof. The temperature of the air near the roof is a difficult one to measure due to the sun, and though the record given by thermocouples was taken, yet the surface coefficient, as well as the transmittance from air to air, are reckoned from the shade temperature. The air temperature near the roof would in practice not be known and the outside temperature spoken of would be the shade. The same applies to walls exposed to the sun.

As is well known the surface action is due to two factors, one the heat transference due to convection which is by no means a simple one, and the other the radiation, which is dependent on the temperature of other surrounding surfaces. Under ordinary room conditions and with thick walls the variations which occur are of little moment, but it shows up in some of the instantaneous values. This is well illustrated by the roof. Its average inner surface coefficient is high because it faces the comparatively hot ceiling below. For instance one afternoon the sun heated the roof so that the heat flow through it gradually decreased until it reached a minimum of no flow in either direction, afterwards gradually rising to an outward flow again, see Fig. 7. During 5 hours of this time the air temperature below the roof was less than that of the under surface, and under these conditions heat should have been flowing from the roof to the attic. However, on account of the direct radiation that factor of the surface action kept the flow in the other direction. The slate side of the roof showed a similar action. After the sun had gone down the air was warmer than the roof due to the latter's rapid radiation to the sky. The values given are therefore good averages for a variety of natural conditions, as in no tests was there a continuation of a uniform type of weather.

As previously stated the flow measured in all tests was the loss from the room into the wall. At any given time this need not be, and is not the same as the flow from the outer surface, but it is one of practical interest and fixes the radiation

required to counteract it at that time. Due to the heat storage capacity of the walls there is always a lag between a change of temperature outside and the corresponding change in the rate of heat loss from the room.

A plotting of a log of the rate of heat loss and the temperature conditions is of interest, and this is the first time that such records have been available. Some such plots are given in the following but to avoid confusing them too much only a few of the temperature curves are shown. In all the curves it will be noted how strongly marked is the effect of a change of the inside temperature on the rate of

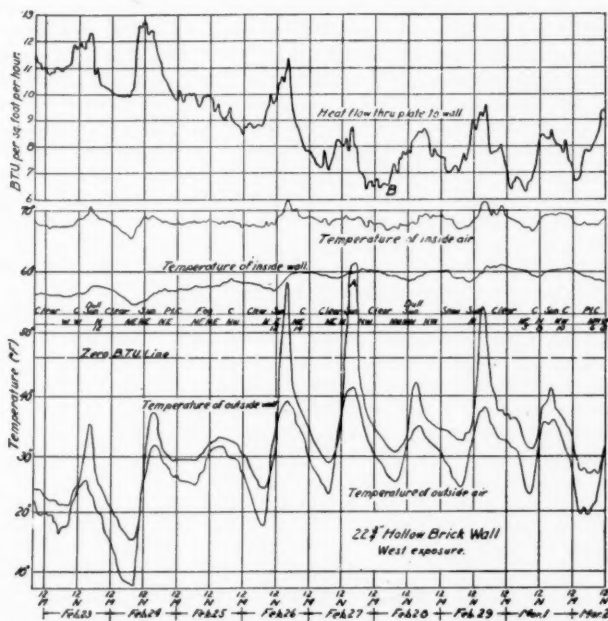


FIG. 4. LOG OF HOLLOW BRICK WALL C, WESTERN EXPOSURE

flow. These tend to mask the more smoothed out variations due to outside temperature changes, although the latter are much the larger.

Fig. 4 shows the log for the 22 in. hollow brick wall, test C, having an open west exposure. The outside air temperature is that given by "shaded" thermocouples, that is protected from the sun by paper shields, although even then they probably register somewhat higher than the true air temperature which itself is indefinite due to the sun heated walls. It will be noted that the outside air and wall temperatures rose most days to sharp peaks, those of the wall being the greater. The corresponding maximum depression in the heat flow occurs about 12 hours later, unfortunately at times when the room temperature rose, but still the effect of these waves can be seen. Thus B is the heat flow peak due to A, temperature rise.

The general constancy of the inside wall surface coefficient is shown by the

constant ratio of the heat flow to the temperature difference between inside air and wall. Considering the outside wall, the similar outside temperature difference shows no constancy and, at some points is negative. At the peaks, such as A, there is the largest difference, but actually at this time there would be a large flow of heat into the wall. Of course because the wall has a higher temperature than the air there must be a flow of heat from it to the air as fixed by the ordinary convection laws. This heat is however not that lost from the room, but is supplied directly by the sun. Thus at these times heat is flowing both ways from the outer surface, into the wall and into the air.

Fig. 5 is for a solid brick wall $8\frac{3}{4}$ in. thick with north exposure. As this did not have the sun directly on it the temperature peaks are not as prominent. The depressions for the heat flow occur about 9 hrs. after the outside rise in temperature. The wall tested was partly protected by other buildings from the north and northwest winds, and the general increasing of the outside surface coefficient by the high west winds on the 17th can be seen by comparing the difference between the two lower curves with the same on the 14th, thus showing that the outer surface coefficient was increased.

Fig. 6 shows a record of a 24 in. concrete basement wall similar to that reported under A. The wall had an east exposure. A meter plate was placed on each face of the wall so as to measure the flow out of as well as into it. The log covers 25 hrs. only. The room conditions and flow into the wall were fairly constant. The flow out of the wall was instructive and illustrates its variability, making up at night for its low values during the day. At 10:25 A.M. the sun came out and immediately the direction of flow was reversed. It appeared intermittently up to 12:15 when, since the wall faced nearly due east, it no longer shone on it. Although the rays were comparatively weak and had a very small inclination to the wall, yet the flow reached a negative value of 14.5 B.t.u., which is greater than its maximum positive one. It will also be seen that during this same time of negative flow, the air temperature remains lower than the wall surface.

Fig. 7 gives the simultaneous values for the ceiling and roof shown in Fig. 3. Only air temperatures are plotted, that for the outside air again being the shade value. The temperature of the loft air is that near the roof, the average value of which was 1.4 deg. fahr. higher than that above the ceiling. The temperature of the room air is that under the ceiling and was of course higher than that at the breathing line. The B.t.u. flow through the ceiling is very irregular due to the difficulty of regulating the radiator with an ordinary globe valve. The influence of the sun on the roof flow has been discussed previously and is clearly seen from the curves. The attic temperature is affected by the wind direction, and the infiltration into it would be higher with a west than with a south wind. Comparing April 2 P.M. with April 3 P.M., the latter with a south wind has the higher temperature.

The results of this test can be used to check up the room radiation. First a comparison will be made between the average room and attic losses during the period of the test.

As shown in Fig. 3 heat flows from the room into the attic, and from there through the roof. The room is a small part of a wing of similar rooms having a common attic. Using B.t.u. loss from Table 1 and dimensions from Fig. 3, the average heat losses per foot run of the wing for the time of the test can be obtained.

As the room was kept at 75 deg. fahr., which was higher than that of the other rooms, the excess found will be greater than the average. There would how-

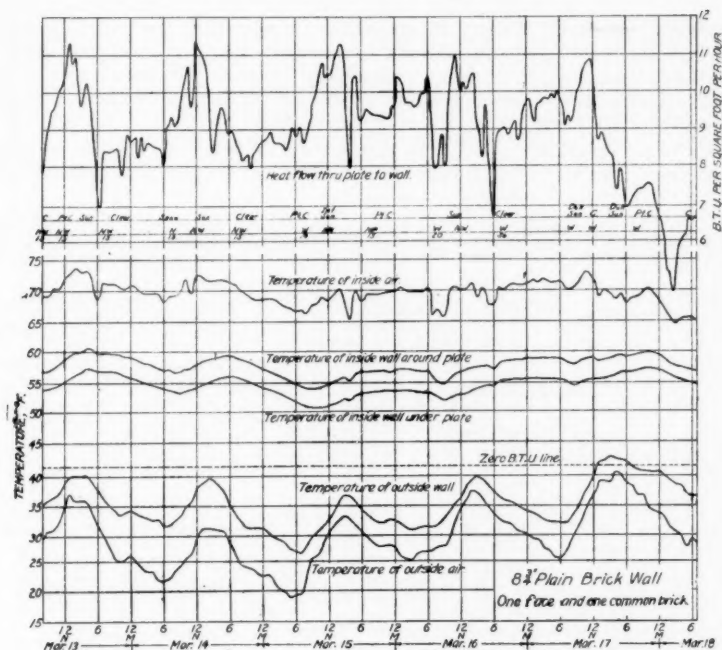


FIG. 5. LOG OF SOLID BRICK WALL D, NORTHERN EXPOSURE

ever be some air changes in the attic due to some vent openings under the eaves.

Considering next the extreme condition of -10 deg. fahr. temperature outside, the corresponding loft temperature can be roughly calculated and will be 40 deg.

Flow through ceiling.....	234 B.t.u. per hr.
Flow through partition wall.....	11 B.t.u. per hr.
Total flow to attic.....	245 B.t.u. per hr.
Flow through roof.....	171 B.t.u. per hr.
Excess to attic.....	74

fahr. The heat losses from the room with 70 deg. inside, neglecting small corrections, are:

Part	Area sq. ft.	Factor	T_d	B.t.u. per hr.
Outside wall 23 in. brick..	115	0.2	80	1850
Glass.....	75	1.1	80	6600
Ceiling.....	300	0.6	30	5500
Partition wall.....	50	0.23	30	345
Total.....				14,195

The radiator is a 23 in., 3 col., 25 sections, which with 215°F steam will give 17,440 B.t.u. per hr., or an excess of 3245 B.t.u. above the total wall losses. As the room has 4300 cu. ft. vol., the B.t.u. required for one air change will be 6200, so that the excess radiation will take care of half a change per hour, which

should be sufficient with the assumption of no wind. Thus the original method of estimation used to fix the radiation agrees closely with what would be needed for the extreme condition. Such an extreme is, however, very rare and of very short duration in the Pittsburgh district, so that the heat capacity of the wall would somewhat reduce the losses at the extreme period.

As the tests reported in this paper were made in order to determine the possible usefulness of the Heat Flow Meters in the heating and ventilating field the following conclusions are suggested:

1. They successfully give values for the desired thermal factors, but for heavy walls such values will be for the average ordinary weather conditions, unless they are

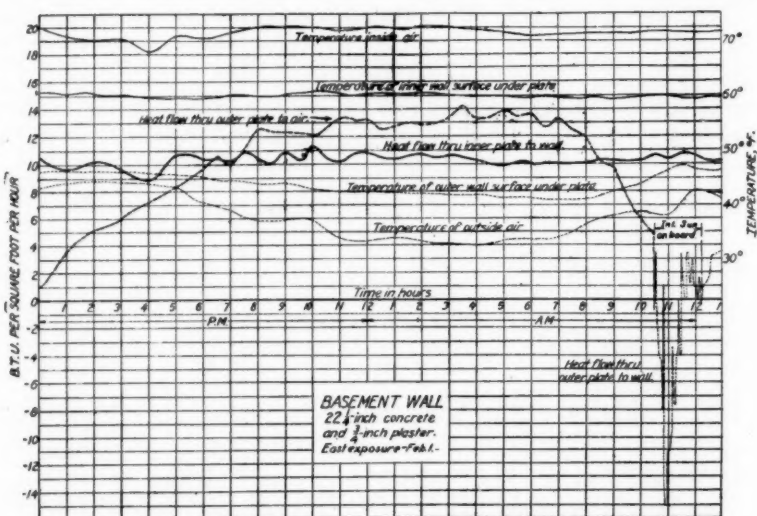


FIG. 6. LOG OF 24 IN. CONCRETE WALL, NORTHERN EXPOSURE, SHOWING FLOWS INTO AND OUT OF THE TWO SURFACES

made in a district where an extreme type will prevail for some days. Thus in only a few instances was it possible to note any effect due to wind, as it never remained constant for long.

2. Independent investigations would be needed for determining the air infiltration through the materials themselves.

3. Their use permits of choosing both a variety of samples of a given construction and their exposure. It obviates the expense of building special samples with their probable superior workmanship and also the time needed for these to dry.

4. They can be used for thicker walls than it is possible to handle as samples, and these can be chosen for any desired age.

5. The time and cost of setting up for a test are small. Even with a full set of temperature readings it is possible to run three or four tests at the same time so that the labor cost per test is less than for similar laboratory tests. With some automatic recording this can be still further reduced.

6. They enable details of the actions to be studied, and their variations traced.

7. Values obtained in this manner are free from the doubt of specially constructed samples and laboratory conditions. The advisability and need of the latter is not

questioned, however, and those made in the manner here outlined should, for a thorough investigation, be in conjunction with others having the various fixed condition that can be obtained in the laboratory.

Acknowledgment is made to the U. S. Bureau of Mines, Pittsburgh, Pa., for their cooperation in giving facilities for the use of the various structural parts, and to G. H. Eisenhart of the Laboratory staff, who assisted in the work here reported.

APPENDIX A

Work on the calibration of the meter plates was not completed at the time of writing the former paper presented to the Society at the January, 1924, meetings. It was

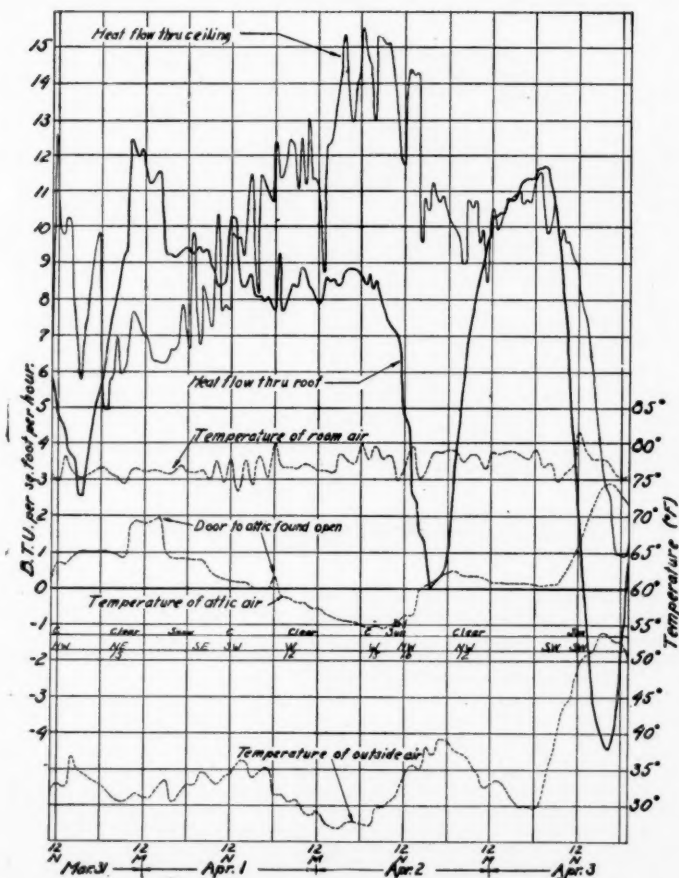


FIG. 7. LOG OF CEILING AND ROOF F AND H

pointed out that there appeared to be a different heat distribution through the thin plates with air contact on one side, to what occurred with close contact. Plates have been tested since then with air contact, thus duplicating a wall application, as shown in Fig. 8.

The general rig up was similar to that described in the previous paper for single direction of heat flow except that one cold plate was omitted so that the last plate had one side exposed to the air. The air was kept at a constant temperature by passing the cooling water through a radiator as shown, and then through the other cold plate. The circulation of the air was by natural convection, and its temperature measured at the lower end by a row of thermocouples in parallel. The constancy of air temperature attainable was very good, and at no time gave trouble; any variation in it of course included that of the cooling water. In one test after approximate constancy of heat flow was approached the air temperature readings over a period of 20 hr. show a maximum variation of 0.2 deg. Fahr.

These tests showed that the $\frac{1}{16}$ in. plates give a 3 per cent lower differential reading

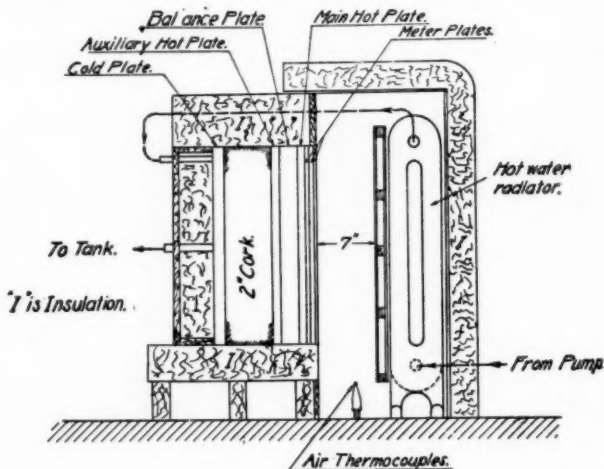


FIG. 8. ARRANGEMENT FOR TESTING METER PLATES WITH ONE FACE AIR EXPOSURE

with air exposure than with close contact, and the $\frac{1}{8}$ in. plates a 2 per cent. These corrections were used in the application tests.

As shown in the previous paper the heat flow into a surface under ordinary conditions is variable due to the natural air currents. In the above tests with steady air conditions such variations were absent or just noticeable.

Although these varying flows approach more nearly to what actually occurs at the surface, yet they make the taking of a differential reading more difficult. To overcome this, and to average and smooth out the variations without increasing the thermal resistance of the plate, one was made with a No. 26 B and S copper plate on its exposed side. This produced the desired result, and even under forced conditions of air flow, its readings changed regularly. Although it has not as yet been proved, it is also believed that this will make the flow distribution the same with any type of contact, and that the smooth finish of the metal plate will more than compensate for the thermal resistance of the copper by reducing the surface resistance. Even with the present plates their low surface resistance as compared with a building wall makes the equivalent thermal resistance added to the wall less than that actually possessed by the plate and air gap.

VALUE OF THE KATA THERMOMETER IN EFFECTIVE TEMPERATURE STUDIES

By MARGARET INGELS,¹ PITTSBURGH, PA.

MEMBER

THE Kata thermometer² is an instrument which was designed to measure the cooling power of air, upon the assumption that the relative comfort of people depends on the cooling power of the air surrounding them. It is a scientific instrument. Effective temperature³ is the measure of the air condition which affects the human senses, and is the resultant of the dry bulb temperature, the wet bulb temperature and the velocity of air. Effective temperatures were established by experiments with human subjects. This report concerns the work of adapting the scientific instrument for measuring effective temperatures.

The Kata thermometer may be used as a *dry* Kata and as a *wet* Kata. The dry Kata loses heat by radiation and convection. The wet Kata loses heat by radiation, convection and evaporation. The dry Kata and wet Kata do not have equal surface temperatures when exposed in the same air, therefore the heat losses by radiation and convection are not equal for the two. It then follows that the difference in the rates of heat loss is not equivalent to heat lost by evaporation on the wet Kata.

The following relations of temperature, air velocity and rate of heat loss are given by Dr. Hill:⁴

Dry Kata

$$\text{Still Air} \quad H = \frac{F}{T} = 0.27 (97.7 - t)$$

$$\text{Velocity below 200 ft. per min. } H = (0.11111 \ 0.01584\sqrt{v}) (97.7 - t)$$

$$\text{Velocity above 200 ft. per min. } H = (0.07222 \ 0.01861\sqrt{v}) (97.7 - t)$$

Wet Kata

$$\text{Velocity below 200 ft. per min. } H = (0.19444 \ 0.08118\sqrt{v}) (97.7 - t)$$

$$\text{Velocity above 200 ft. per min. } H = (0.05556 \ 0.10505\sqrt{v}) (97.7 - t)$$

¹ Research Head, A. S. H.-V. E. Research Laboratory.

² The Science of Ventilation and Open Air Treatment, by Leonard Hill, London, 1919.

³ Equal Comfort Lines, by F. C. Houghten and C. P. Yagloglou, AMERICAN SOCIETY HEATING AND VENTILATING ENGINEERS' JOURNAL, March 1923.

⁴ Special Report Series No. 73, Medical Research Council The Kata-thermometer in Studies of Body Heat and Efficiency.

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Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Kansas City, Mo., June 1924.

H = millicalories per sq. cm. per second.

F = *Kata Factor* = millicalories lost by Kata in dropping from 100 deg. fahr. to 95 deg. fahr. per sq. cm.

T = time of drop in seconds.

t = temperature in fahrenheit degrees, dry bulb temperature for dry Kata formulae and wet bulb temperature for wet Kata formulae.

v = velocity of air in feet per minute.

The work done by this Laboratory⁵ on the Kata thermometer determined the relation of temperature, velocity and rate of heat loss to be as follows:

Dry Kata

$$H = \frac{\text{Kata Factor} \times 0.003687 \times 3600}{T}$$

$$H = \text{B.t.u. M.S.H.} \frac{b}{15.9 \ 0.0386v} + 8.25$$

Wet Kata

$$H = \text{B.t.u. M.T.H.} \frac{b}{19.86 \ 0.0772v} + 5.34$$

H = B.t.u. per sq. ft. per hour.

T = time of cooling in seconds.

B.t.u. M.S.H. = Mean Sensible Heat Difference between surrounding air and air at mean temperature of Kata

B.t.u. M.T.H. = Mean Total Heat Difference between surrounding air and air at mean temperature of Kata.

v = velocity of air in feet per minute.

Substituting values in both sets of equations the formulae for the dry Kata check each other. The formulae for the wet Kata do not check. Tests were made in the constant temperature rooms of this Laboratory and their results proved the correctness of those formulae first set up by this Laboratory.

In further consideration only the second set of formulae will be used.

The rate of cooling of the dry Kata depends, first, on the mean between the sensible heat of the surrounding air and the sensible heat of air at mean Kata temperature, and second on the velocity of the air. As sensible heat is a function of the dry bulb temperature, the rate of cooling of the dry Kata is a function of the dry bulb temperature and velocity.

The rate of cooling of the wet Kata depends, first, on the mean between the total heat of the surrounding air and the total heat of air at mean Kata temperature, and second on the velocity of the air. Total heat is a function of the wet bulb temperature, therefore the rate of cooling of the wet Kata is a function of the wet bulb temperature and velocity.

Effective temperatures approach the dry bulb at low temperatures and the wet bulb at high temperatures. In still air the effective temperature is equal to the dry bulb at 32 deg. regardless of the wet bulb. That is, the dry Kata approaches a measure of the effective temperature at low temperatures, likewise the wet Kata approaches a measure of the effective temperature at high temperatures. People react in one extreme as a dry Kata and the other extreme as a wet Kata.

From the table, which was compiled from the cooling rates of dry and wet Katas and the effective temperature charts, it can be seen that the air conditions giving equal comfort do not give equal rates of cooling on either of the Katas. Therefore the cooling rates on either or both Katas will not be an index to comfort.

⁵ Temperature, Humidity and Air Motion Effects in Ventilation, by O. W. Armspach and Margaret Ingels, AMERICAN SOCIETY HEATING AND VENTILATING ENGINEERS' JOURNAL, March 1922.

It is evident that the Kata is not a complete instrument in itself for measuring effective temperatures. It does have a use in this work and that is as an anemometer.

TABLE 1. SHOWING THAT DIFFERENT POINTS ON THE SAME EFFECTIVE TEMPERATURE LINES DO NOT GIVE EQUAL COOLING RATES ON EITHER THE DRY OR WET KATA

Effective temperature degrees	Given conditions		Velocity ft. per min.	Cooling	
	D. B. degrees	W. B. degrees			
30	30	30	0	135	286
30	29	20	0	136.5	306
60	60	60	0	73.6	208
60	71	44	0	52.6	255
70	70	70	0	54.6	167
70	86	55	0	22.7	224
105	105	105	0	-14.3	-66.8
105	163.5	97.5	0	-90.5	0
30	58	58	500	214.5	133
30	52	34	500	248	213.5
70	81.2	81.2	500	87.54	54.2
70	91	58.4	500	33	131.0

eter. The relations of temperature, rate of cooling off the Kata, and air velocity are given by the equation, the temperature and rate of cooling are easily measured so the velocity can be figured. Either the dry Kata or wet Kata may be used for determining velocities. The time of cooling for the dry Kata is longer, and variables introducing errors do not enter in the dry Kata readings as much as in wet Kata readings, therefore the dry Kata can be used more accurately as an anemometer than the wet Kata. The Kata is of value only as an anemometer in measuring effective temperatures and in future work will be used fundamentally as such.

The three variables on which effective temperatures depend are dry bulb temperature, wet bulb temperature, and velocity. Effective temperatures have been established through a range of 30 to 115 deg. This means a dry bulb temperature range of 29 to 170 deg., wet bulb of 20 to 115 deg., and velocity range from still air to 500 ft. per min. For each value of one of the variables, each other variable can vary through its entire range. This means an infinite number of combinations.

Charts have been made to simplify determining the effective temperatures for any air condition. The dry bulb and wet bulb temperatures may be read from a sling psychrometer. The rate of cooling of the Kata is found as follows: The dry Kata loses a known amount of heat in dropping in temperature from 100 to 95 deg. This amount is written on the back of the thermometer and is given in millicalories per square centimeter of Kata surface. It is known as the *Kata factor*. The Kata factor is changed to B.t.u. per square foot by multiplying by 0.003687. The time is taken for the Kata to cool from 100 to 95 deg., and the rate of cooling in B.t.u. per square foot per hour.

$$H = \frac{\text{Kata Factor} \times 0.003687 \times 3600}{\text{Time of Cooling in Sec.}}$$

$$H = \frac{\text{Kata Factor} \times 13.273}{\text{Time of Cooling in Sec.}}$$

With the dry bulb and wet bulb temperatures and the rate of cooling of the Kata the effective temperature is found as follows: Refer to the chart for the known dry bulb and on this chart find the intersection of the wet bulb temperature and the rate of cooling of the dry Kata, and read the effective temperature for this point.

An additional scale of velocity of air in feet per minute is given, for if the velocity is known the effective temperature can be read direct without knowing the cooling rate of the Kata.

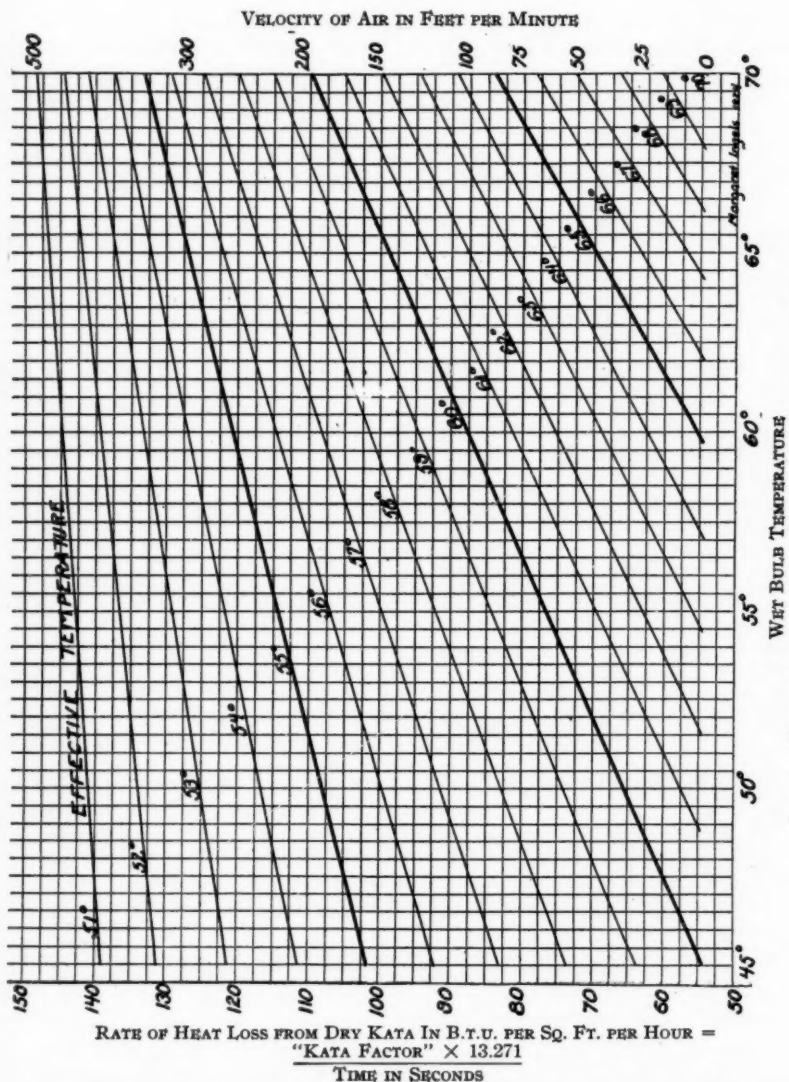


FIG. 1. CHART FOR OBTAINING EFFECTIVE TEMPERATURE READING

Effective temperatures can be obtained from such charts with the simplest operation and explanation, and combinations of the variables affecting effective temperatures are given.

The dry bulb chart of 70 deg., Fig. 1, accompanies this report, and shows the characteristics of all the charts which have been prepared.

CORRELATION OF SKIN TEMPERATURES AND PHYSIOLOGICAL REACTIONS

By W. J. McCONNELL, M. D.¹ (NON-MEMBER) AND C. P. YAGLOGLU,² (MEMBER)

PITTSBURGH, PA.

Accurate Measuring of Skin Temperature

THE physiological significance of skin temperatures has attracted less study than it probably deserves on account of the technical difficulties in accurately measuring the temperature of surfaces of the body.

It is obvious that temperatures registered by a thermometer whose bulb is only partly in contact with the body are resultants of skin and environmental temperatures. If the exposed part of the bulb is covered in an effort to reduce the influence of external conditions, the heat loss from the covered portion of the skin is materially affected, and the temperature indicated by the thermometer is higher than the true temperature of the surface.

Many devices have been invented to eliminate the influence of environmental temperature. Thus Davy³ used a thermometer whose bulb in form was nearly cylindrical and fixed to a small piece of cork made hollow and lined with fine wool.

The New York State Commission on Ventilation⁴ in certain experiments measured the temperature of the skin covering the chest by means of accurate long stemmed chemical thermometers, the bulbs of which were covered by stiff paper, so that an air pocket was left between the bulb and the paper, and thus the bulb could not touch the skin. The thermometer was at first warmed to a temperature of 29.4 deg. cent. (85 deg. fahr.), and was then placed well within the clothing and was allowed to remain there for a period of 10 min. before the reading was taken.

Benedict, Miles, and Johnson⁵ used the electrical method which consists of two copper-constantan junctions, one immersed in a constant temperature bath a Dewar flask, and the other applied to the skin. The current set up due to the difference in temperature between the two junctions was measured by the deflection obtained on a galvanometer. The junction was backed with a light tuft

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³ Davy, John, *Philosophical Transactions* of the Royal Society of London, Feb. 17, 1814, p. 590.

⁴ Ventilation, Report of the New York State Commission 1923, p. 39.

⁵ Benedict, Miles, and Johnson, *Proceedings of National Academy of Science*, 1919, p. 218.

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Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Kansas City, Mo., June 1924.

of cotton batting and installed rigidly in a piece of hard rubber. The junction in the bath was kept at 31 and 32 deg. cent.

In the studies of the physiological reactions to high temperatures and humidities conducted by the U. S. Bureau of Mines, Pittsburgh, Pa. in cooperation with the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, surface temperatures have been recorded in the majority of experiments, and the methods used and results obtained are the subject of this paper.

When the work was first started early in 1922, specially designed flat bulb thermometers were constructed and used for measuring surface temperatures. On account of the inconsistent results obtained, this method was soon replaced by the thermoelectric method described in a previous article.⁶ The methods described there have been improved and the thermocouple finally adopted is shown in Fig. 1.

Thermocouple for Determining Skin Temperature

A glass tube encases the copper-constantan thermocouple wires which are securely fixed at the two ends of the tube by means of sealing wax. The junction to be applied to the surface is exposed so as to be in contact with the skin, the

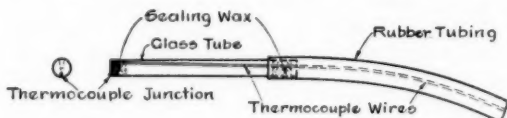


FIG. 1. THERMOCOUPLE USED IN DETERMINING SKIN TEMPERATURE

wires leading to the junction for a length of $\frac{3}{16}$ in. are bent over so as to run parallel to the surface but covered with a very thin layer of sealing wax. The cold junction is kept at a constant temperature of 32 deg. fahr. in a bath of melting ice. The electromotive force set up due to the difference in temperature between the two junctions of the system is read on a Tinsley potentiometer and converted into degrees fahrenheit by means of a calibration curve. To obtain the surface temperature of any part of the body the glass tube is placed at a right angle in firm contact with the skin and held in this position approximately a minute until a constant reading is obtained on the potentiometer. In atmospheric temperatures considerably lower or higher than that of the body the glass tube is kept in warm water of about body temperature in order to reduce the time for making the observations. The heat capacity of this device is rather small when compared with the capacity of a clinical thermometer, and there is little interference with the heat loss from the portion of the skin where the thermocouple junction is applied, as the diameter of the glass tube is less than $\frac{1}{4}$ in.

The procedure followed in conducting the tests can be found in a previous study.⁷ The first surface temperature reading was made at the expiration of the rest period before entering the test chamber. The temperatures of the cheek,

⁶ Body Temperatures and Their Measurements, by W. J. McConnell and F. C. Houghten, JOURNAL A. S. H. & V. E., July 1922.

⁷ Some Physiological Reactions for High Temperatures and Humidities, by W. J. McConnell, and F. C. Houghten, JOURNAL A. S. H. & V. E., March, 1923, pp. 131-164.

abdomen (just above the umbilicus), and dorsum (at the waist line) were recorded. Readings were similarly taken at intervals during the tests.

The results obtained by Davy,⁸ Benedict, Miles and Johnson⁹ show extraordinary differences in skin temperatures at different points. Davy concludes that the temperature of parts diminishes as the distance of parts from the heart increases. The New York Commission¹⁰ found that the temperature of the skin on the chest, recorded in their experiments, conformed fairly well with those of Davy, Kunkel,¹¹ Stewart¹² and Benedict, Miles and Johnson.¹³

Relation of Skin Temperature to Thermal Properties of Body

It is well known and has been emphasized by Pembrey and Collis¹⁴ that the skin becomes flushed and warm when the body is exposed to warm moist air, but

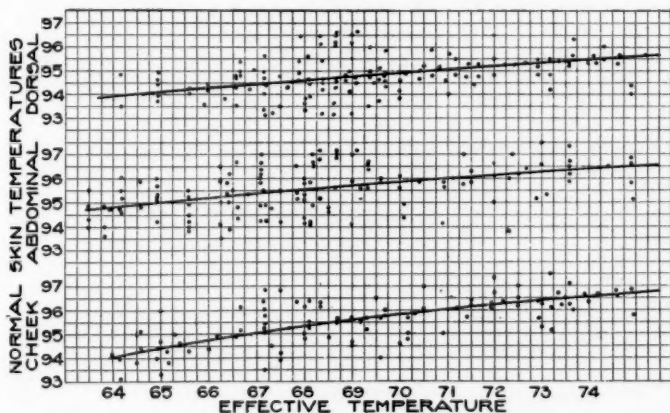


FIG. 2. RELATION BETWEEN EFFECTIVE TEMPERATURE AND SKIN TEMPERATURE

the relation between skin temperatures and other physiological reactions have not to our knowledge been studied, and the importance of skin temperatures to the thermal properties of the human body is not generally recognized. When the subjects were lightly dressed, wearing trousers and shirts, and following a 2 hour rest period in a well ventilated room, but not accurately controlled, the cheek, abdominal and dorsal temperatures were taken, and Fig. 2 shows the average surface temperature of five subjects plotted against effective temperature.¹⁵

The humidity in the room where these readings were taken varied from 35 to 70 per cent on different days, while the temperature conditions were not constant

⁸ Work cited.

⁹ Work cited.

¹⁰ Work cited.

¹¹ Kunkel, *Zeitschrift für Biologie*, 1889, XXV, p. 55.

¹² Stewart, *Studies of the Physiological Laboratory*, Owens College, Manchester, 1891, V. P. 100.

¹³ Work cited.

¹⁴ Pembrey, M.S., and Collis, E.L., Appendix 111-2nd, Report of the Departmental Committee on Humidity and Ventilation in Cotton Weaving Sheds, London, 1911.

¹⁵ Determining Equal Comfort Lines for Still Air Conditions, F. C. Houghten and C. P. Yagloglou

JOURNAL A. S. H. & V. E., March 1923.

which probably accounts for the variation in the experimental points. Other factors, such as previous activities and changes in the clothing worn from day to day may add their influence.

The cheek temperature varies from 94 deg. fahr. at 64 deg. effective temperature to about 97 deg. at 75 deg. effective temperature, while the variation in the abdominal and dorsal temperature is less than 3 deg. This indicates that the cheek being exposed to the air is considerably more affected by external conditions than the protected parts of the body. It is also noted that at comparatively low tem-

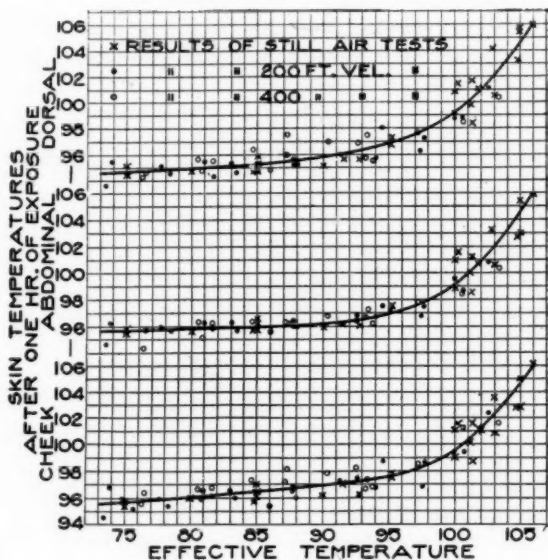


FIG. 3. RELATION BETWEEN EFFECTIVE TEMPERATURE OF ENVIRONMENT AND SKIN TEMPERATURE AFTER 1 HOUR'S EXPOSURE

peratures the abdominal temperature is about one deg. higher than that of the cheek, while the dorsal temperature is the lowest of all.

The parallelism in the two upper curves shows that the clothed parts of the body are similarly affected by external conditions. At about 70 deg. effective temperature the cheek temperature approaches that of the abdomen, and at 75 deg. effective temperature the two are identical. The curves are not straight lines due to the variable amount of perspiration available for evaporation at the different temperature conditions, as a result of which the curves fall off with increasing temperature.

Before entering the test chamber the subjects stripped to the waist. In the chamber the temperature conditions were maintained constant and Fig. 3 shows

the results of surface readings after one hour's exposure to the temperature in the chamber.

Tests in Still Air and Moving Air

The crosses represent tests conducted in still air while the points and circles represent tests conducted in air moving at the rate of 200 and 400 ft. per min. respectively. This chart indicates that the surface temperatures follow very closely the scale of effective temperature irrespective of the velocity of the air.

On exposing the unclothed body to an unvarying atmospheric temperature and humidity it was found that the temperature of the skin attained a constant value after a period of exposure depending upon the severity of the test. This is illustrated in Fig. 4.

Here the temperature conditions were maintained at 105 deg. fahr. with 100 per cent relative humidity. The average surface temperatures for all subjects

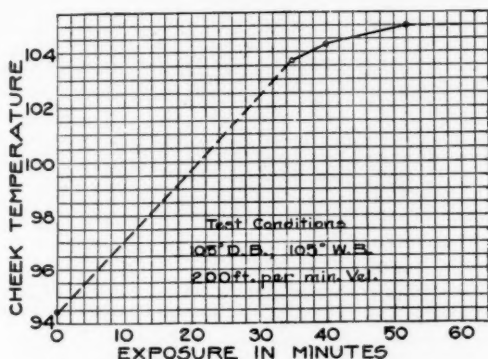


FIG. 4. RELATION BETWEEN CHEEK TEMPERATURE AND TIME OF EXPOSURE TO A SATURATED ATMOSPHERE OF 105 DEG. AND 200 FT. VELOCITY

were 94.4, 94.1, and 95.0 deg. for cheek, abdomen and dorsum, respectively. After exposure in the test chamber for 35 min. these values became 103.7 deg. fahr. for all three surfaces. The average three surface temperatures of two subjects who remained in the chamber for 50 min. reached 105.0 deg. fahr. showing an increase of only 0.7 deg. This indicates that the surface temperatures under those conditions do not exceed the temperature of the air.

The values given in Figs. 3 and 4 and in the accompanying charts are based on changes taking place after 1 hour of exposure to the test conditions. In the very high-temperature tests, where the subjects were unable to remain an hour, extrapolation would lead to erroneous results. Therefore our data is limited to 106 deg. fahr. surface temperature which is approximately the highest value actually obtained in the tests. When the time of exposure was less than 1 hour, but not less than 50 min., a straight-line extrapolation gives reasonable results. In tests exceeding 106 deg. effective temperature, where the period of exposure was

much shorter than 50 min., the values obtained by extrapolation of surface temperatures were materially higher than those of the dry-bulb temperature readings of the air and therefore are not included in this study.

It will be observed in Fig. 3 that for low temperature the rise in the surface temperature is not great. After 96 deg. effective temperature a gradually increasing rate is manifested, and above body temperature the change is considerable. A comparison of the three curves shows that at low temperatures the abdominal and cheek temperatures are practically the same, while the dorsal temperature is the lowest of all. Above body temperature all three readings become identical and approach the effective temperature of the air, until at 106 deg. effective temperature the surface of the body attains the temperature of the environment. It should be kept in mind that the curves do not necessarily bear a definite relation

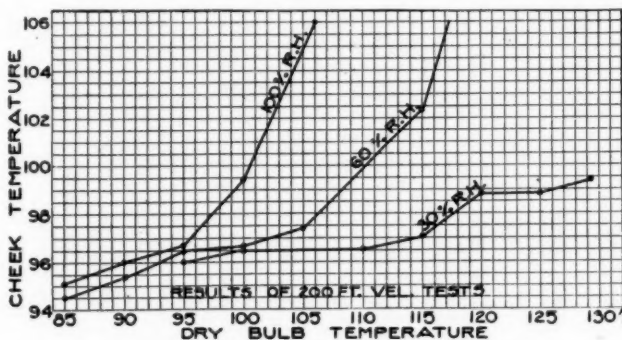


FIG. 5. RELATION BETWEEN CHEEK TEMPERATURE AND DRY-BULB TEMPERATURE WITH 30 %, 60% AND 100% RELATIVE HUMIDITY

to those attained in the primary room due to the difference in the exposed parts of the body in the two cases.

No detailed study could be made of the effect of wind on the surface temperature, due to the fact that complete data is not available for the entire range of temperatures employed in the tests with still air, and with 400 ft. per min. velocity. However, indications show that air motion at low temperatures exerts an appreciable lowering in the temperature of the body surface, while at temperatures above that of the body it tends to change the surface temperature to that of the atmospheric environment.

Effect of Humidity on Skin Temperature

The relative humidity in the different tests varied from 15 to 100 per cent but only its effect on surface temperature is shown as indicated in Fig. 5, for constant relative humidities of 30, 60 and 100 per cent and for a velocity of 200 ft. per min.

Attention is invited to the sudden change in cheek temperature occurring at 95 deg. dry bulb for 100 per cent relative humidity, while for 60 and 30 per cent it takes place at 105 and 115 deg. dry bulb, respectively. For saturated temperatures above 100 deg. the cheek temperature and dry-bulb temperature are the same, since all three surface temperature measurements are practically the same,

the cheek temperature as the average representative of the others is used in the correlation study.

The surface temperature, as represented by the cheek, has been plotted against the pulse rate and rectal temperature in Fig. 6 in order to show the correlation between surface temperature and these other two important physiological reactions.

Reference to the curves for body temperatures and pulse rates published in various issues of the JOURNAL¹⁸ OF THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS will disclose that they compare favorably with the plots given here for surface temperature. These charts show how directly the rectal

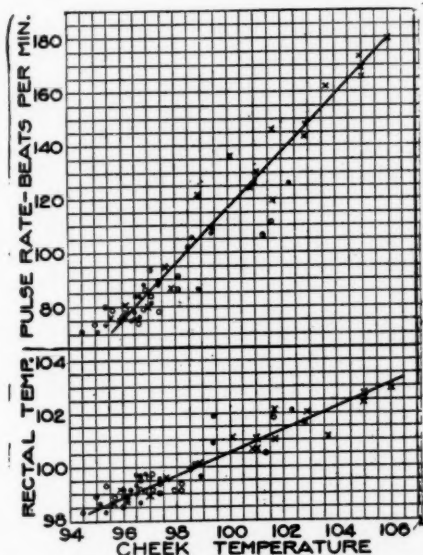


FIG. 6. RELATION OF CHEEK TEMPERATURE TO BODY TEMPERATURE AND PULSE RATE

temperature and pulse rate vary with the cheek temperature. At the normal body temperature the average cheek temperature of the five subjects of the experiments was about 96.0 deg. fahr. while at a body temperature as measured per rectum of 103 deg., the corresponding cheek temperature rose to 106 deg. Similarly when the pulse rate was about normal, the cheek temperature was around 96 deg. fahr., and as the pulse rate increased to 180 pulsations per min. the cheek temperature attained a value of 106 deg. fahr. The difference in slope of the two lines indicates that the pulse rate is a more sensitive index than rectal temperature of the severity of the external conditions.

Enough data are not available at present to afford a study of the thermal prop-

¹⁸ Some Physiological Reactions to High Temperatures and Humidity, by W. J. McConnell and F. C. Houghten, JOURNAL A. S. H. & V. E., March, 1923, pp. 131-163. Further Study of Physiological Reactions, by W. J. McConnell, F. C. Houghten and F. M. Phillips, JOURNAL A. S. H. & V. E., Sept., 1923, pp. 507-514. Air Motion—High Temperatures and Various Humidities Reactions on Human Beings, by W. J. McConnell, F. C. Houghten and C. P. Yagloglou, JOURNAL A. S. H. & V. E., March, 1924, p. 199.

erties of the human body when exposed to freezing conditions. The lowest surface temperature recorded was about 70 deg. fahr. and occurred in a saturated temperature of about 40 deg. fahr. with 400 ft. per min. velocity.

Conclusions

The experimental evidence herein presented indicates that the surface temperature of the human body is directly affected by external atmospheric conditions, and in turn initiates the other physiological reactions. These results likewise emphasize the importance of primary skin reactions upon which effective temperatures are based.

AIR LEAKAGE AROUND WINDOW OPENINGS

By C. C. SCHRADER,¹ PITTSBURGH, PA.

MEMBER

THIS report is the second one dealing with the air leakage problem being investigated at the Research Laboratory. The first report, Air Leakage through the Openings in Buildings, was published in the February, 1924 JOURNAL and included a description of the apparatus and the method of procedure. The tests in both experiments followed the same general routine, and in each case the same frame was used with the same kind of sash, namely, double hung wooden sash, 2 ft. 8 in. by 5 ft. 2 in. by $1\frac{3}{8}$ in.

The principal facts brought out in the first report were that increasing the crack around the perimeter of a plain sash did not materially increase the leakage, and that weather-stripped sash, while permitting much less leakage, showed a small increase in leakage with increase in crack. These facts were established by making several hundred tests. The present report deals with the effect of increasing the width of the stile, that is, increasing the clearance.

Fig. 1 illustrates what is meant by crack and clearance. The crack around the sash perimeter is equal to one half the difference between the width of the frame and the width of the sash, that is, the crack is the same on each side of the sash. The clearance is the difference between the width of the stile and the thickness of the sash. These terms are chosen arbitrarily to distinguish the two principal air passages which are found in double hung windows, and they will be used frequently throughout the report and should not be confused.

In fitting the sash for these tests it was discovered that they were thicker than those used in the first tests, which led to an investigation as to the actual thickness of the sash in each case. All of the sashes were measured with calipers, and it was found that those used for the tests in the first report were $1\frac{21}{64}$ in. thick while those used in the new tests were $1\frac{3}{8}$ in. thick. The stile previously measured was $1\frac{7}{16}$ in. in width. In view of this information a statement made in the first report must be corrected, which was that the difference between the width of the stile and the thickness of the sash was $\frac{1}{16}$ in. when it was actually $\frac{7}{64}$ in. This difference in clearance is probably due to manufacturing practices, as some mills use the specified dimension for the thickness of the sash while others use it for the width of the stile.

¹ Research Engineer, A.S.H. & V.E. Laboratory.
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Presented at the Semi-Annual meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Kansas City, Mo., June, 1924.

In order to test the effect of different clearances it was necessary to have stops that could be moved to any desired distance from the parting bead. Therefore the stops were removed and slots cut in them so that they could be screwed to the frame to give a maximum of $\frac{1}{4}$ in. clearance. Known clearances were obtained by strips of cold rolled steel of convenient thicknesses, and these strips were measured with a micrometer to determine their exact dimensions. A strip of a thickness equal to the desired clearance was inserted between the sash and the stop, the whole pressed firmly against the parting bead, and the stop screwed on in position. The strip was then removed, and in this manner any desired clearance was obtained. When weather stripping was used it held the sash against the part-

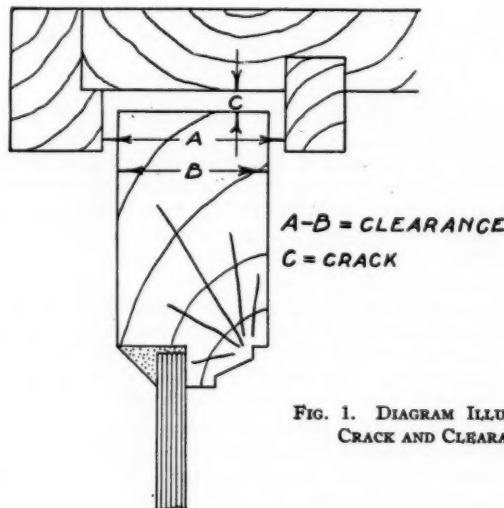


FIG. 1. DIAGRAM ILLUSTRATING CRACK AND CLEARANCE

ing bead, so that the clearances could be obtained in the same manner. The cracks around the stop were sealed to prevent leakage through them.

Procedure

Four sets of sash were fitted with cracks of $\frac{1}{16}$, $\frac{1}{8}$, $\frac{3}{16}$ and $\frac{1}{4}$ in. Each set was tested with clearances varying from $\frac{1}{32}$ to $\frac{1}{4}$ in. Each test was repeated a number of times because no two tests gave exactly the same leakage, and it was necessary to obtain average results. Before duplicating any test the window was opened and closed, and the stops were removed and then returned to as nearly the same position as possible. The weather-stripped sashes were tested in the same way.

Plain Window

Fig. 2 gives the results of tests of a plain window with various clearances. The tests proved that the size of the crack around the perimeter of the sash has no

appreciable effect on the leakage. Therefore the results apply to any window of the type tested with a crack of from $\frac{1}{16}$ to $\frac{1}{4}$ in. In practice most new sashes are fitted with the crack at least $\frac{1}{16}$ in., and this crack becomes greater as the sash dries out and shrinks. It should be clearly understood that each curve is the average obtained from a number of tests, and the results of any one test may vary from the given curve by four or five per cent. The figure shows that the leakage increases rapidly with increase in clearance.

Fig. 3 is obtained from curve 4 in Fig. 2 and is drawn to show the relation of leakage to wind velocity. The graph thus drawn shows a slight double curve, but it is seen that it varies but little from a straight line. In some cases the double

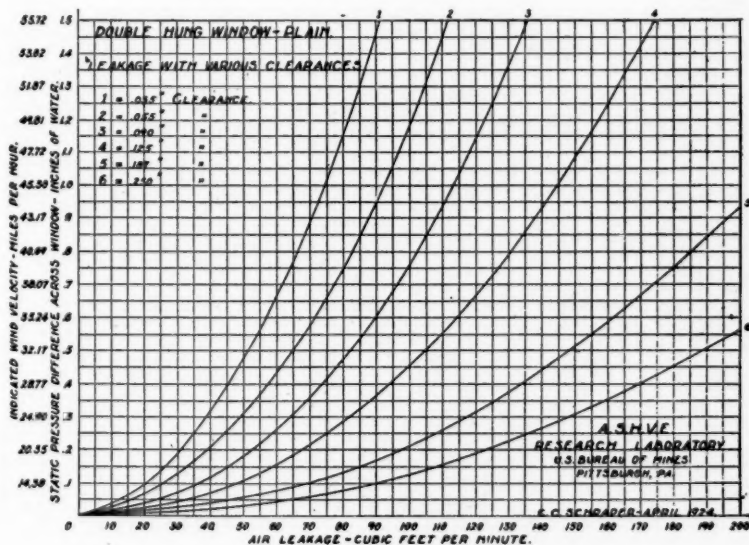


FIG. 2. LEAKAGE THROUGH PLAIN WINDOW WITH VARIOUS CLEARANCES

curve is more pronounced but the curve does not reverse until the velocity has reached twenty-five or thirty miles per hour. Hence no great error is introduced by taking any point on the leakage curve below thirty miles per hour, to determine the leakage per mile of wind velocity.

Since the velocity used in most calculations of window leakage is about fifteen miles per hour, or approximately $\frac{1}{10}$ in. water pressure, values at these points were taken from the curves in Fig. 2 and plotted against clearance. Fig. 4 was thus obtained. It is seen that a straight line passes close to all these points, and that the variation from the points plotted will be within the limits of the above variation of the original tests. The line drawn does not pass through zero leakage for zero clearance, and therefore the leakage will not vary in direct proportion to the clearance, which is to be expected because the surfaces of the window are not

perfectly true, and regardless of how closely they are held together some leakage will always occur. If the sashes were not free to move with the pressure of the wind and were always held in the middle of the stile, a direct ratio of leakage to clearance would be expected. However the movement of the sash introduces a variable element, and it is difficult to determine its effect. It must be remembered also that the leakage at zero clearance is not the elsewhere leakage. The conditions for determining the elsewhere leakage require that certain openings around the sash be sealed so that no leakage will take place through them, whereas regardless

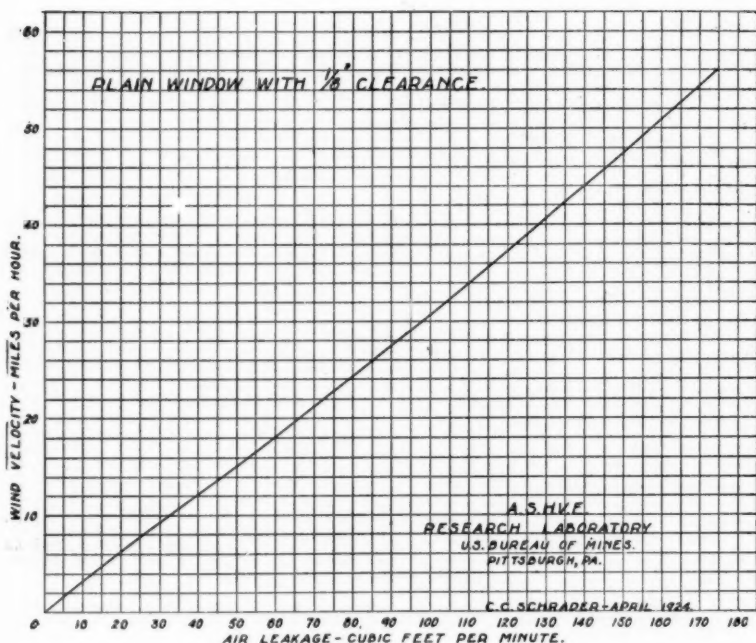


FIG. 3. RELATION OF LEAKAGE TO WIND VELOCITY

of how tightly the sash fits between the stops some leakage will occur through these openings due to the uneven surfaces caused by warping and so forth. (For determination of elsewhere leakage refer to first report mentioned above.) It is probable that the elsewhere leakage, particularly through the pulley holes, increases as the clearance is made greater, but the increase is too small to be differentiated from the variation that may take place in two tests of the same condition.

From the standpoint of the heating engineer the most convenient terms expressing infiltration are cubic feet per hour, per foot of crack, or per mile of wind velocity. Using these terms the infiltration for any room can be easily computed

by multiplying by a constant depending on the wind velocity and the number of feet of crack. Fig. 5 was obtained by expressing the values in Fig. 4 in these terms.

Weatherstripped Windows

In the first report it was shown that the leakage through a weather-stripped window increased as the crack was made larger. Fig. 6 shows what happens when the crack is kept constant and the clearance increased in a window with the interlocking type strip applied. The increase in leakage with increase in clearance is probably

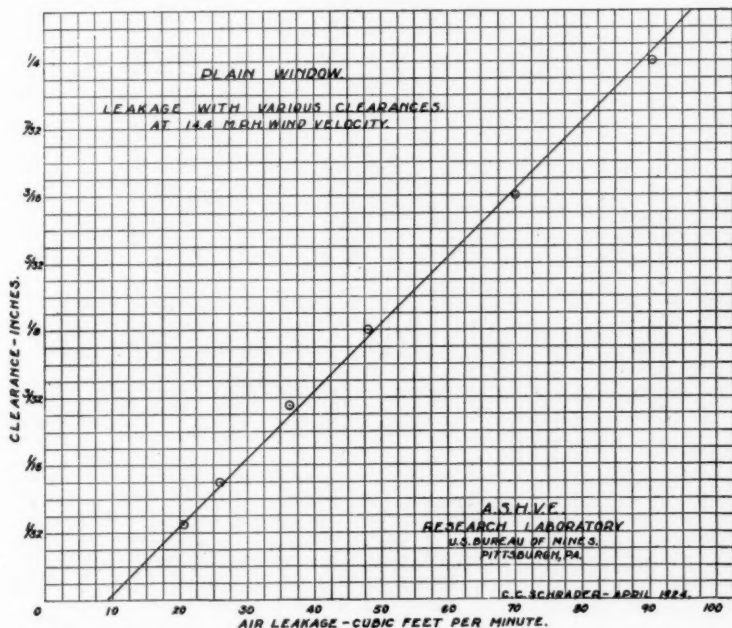


FIG. 4. RELATION OF TOTAL LEAKAGE TO CLEARANCE. PLAIN WINDOW

due to greater leakage through the pulley holes which is caused by decreased resistance in the path of the air to the pulley holes. This evidence supports further the statement previously made that the elsewhere leakage increases slightly as the clearance decreases. In fact, it is the only explanation that can be found for the increase in leakage since the sash is held against the parting bead in the same position by the weatherstripping, all other openings around the sash remaining the same. The increase in leakage is quite pronounced until the clearance becomes equal to the crack, after which the increase is not so rapid. Fig. 7 illustrating this point shows the leakage through windows with the interlocking type

weatherstrip for all the cracks and clearances tested. Fig. 8 gives similar data for windows fitted with rib type weatherstrip. A comparison of the two figures brings out the greater consistency in results obtained with the interlocking type of strip, which is to be expected, as this type of weatherstrip holds the sash in the same position at all times. With the rib strip the sashes are free to move from

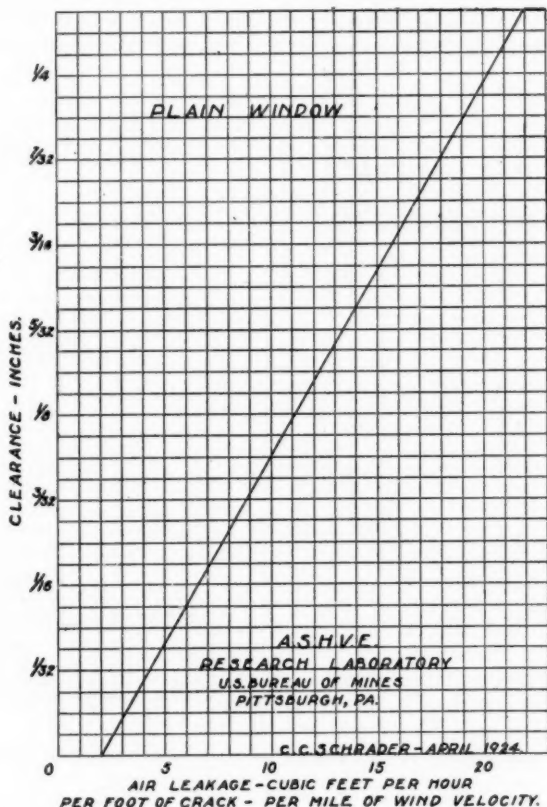


FIG. 5. RELATION OF LEAKAGE PER FOOT TO CLEARANCE.
PLAIN WINDOW

side to side a distance depending on the size of the crack, and since it is impossible to return them to exactly the same position after each test the results become more varied. This change of position of the sash often causes the leakage with a small clearance to be greater than that with a larger clearance, for a slight change is sufficient to affect the difference. This fact is particularly true when the crack

is large and permits considerable divergence from the previous position of the sash.

It is not the purpose of this report to draw a comparison between the relative values of different types of weatherstripping. However, some explanation is necessary to understand thoroughly the conditions under which these tests were conducted so that comparison may be made with other data on the subject. Concerning the interlocking type of weatherstrip all that needs to be mentioned is

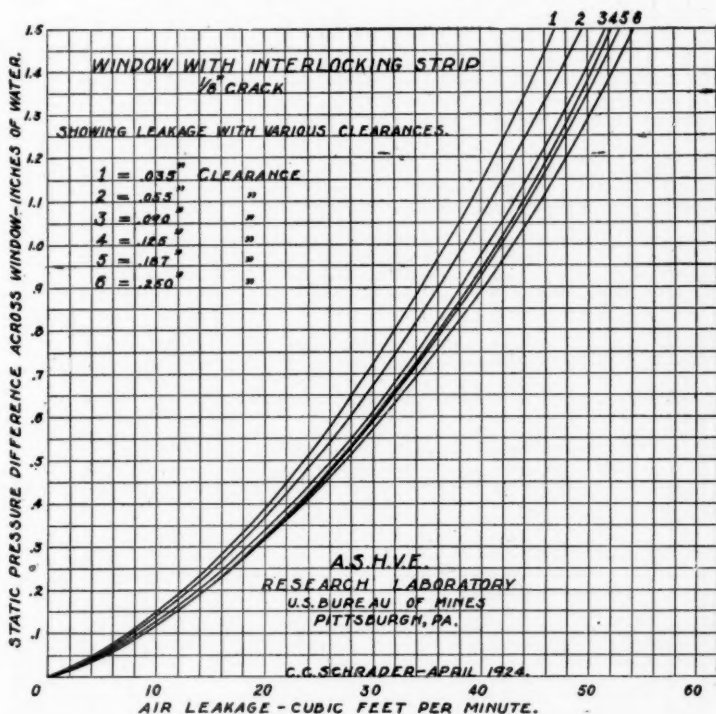


FIG. 6. LEAKAGE THROUGH WEATHERSTRIPPED WINDOW WITH VARIOUS CLEARANCES

that it has metal members on both the sash and the frame, and they are so constructed that one fits into the other. The rib type strip has one metal member fastened on the frame which fits into a groove ploughed in the sash by the carpenter applying the strip. The width of this groove in comparison with the width of the metal member may be the determining factor in the leakage through a window to which this type of strip is applied. In this instance the rib was $1/8$ in. wide and the groove $1/32$ in. wider. De Volson Wood (*A.S.M.E. Transactions*, Vol. 10) gives the lateral expansion of white pine as 2.6 per cent from dryness to saturation.

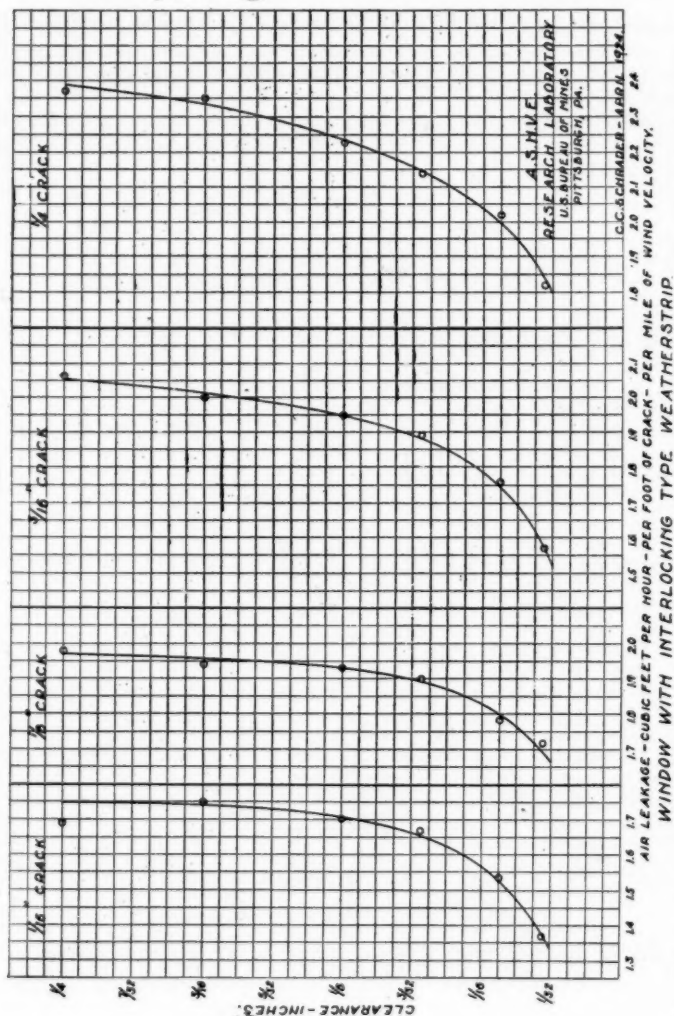
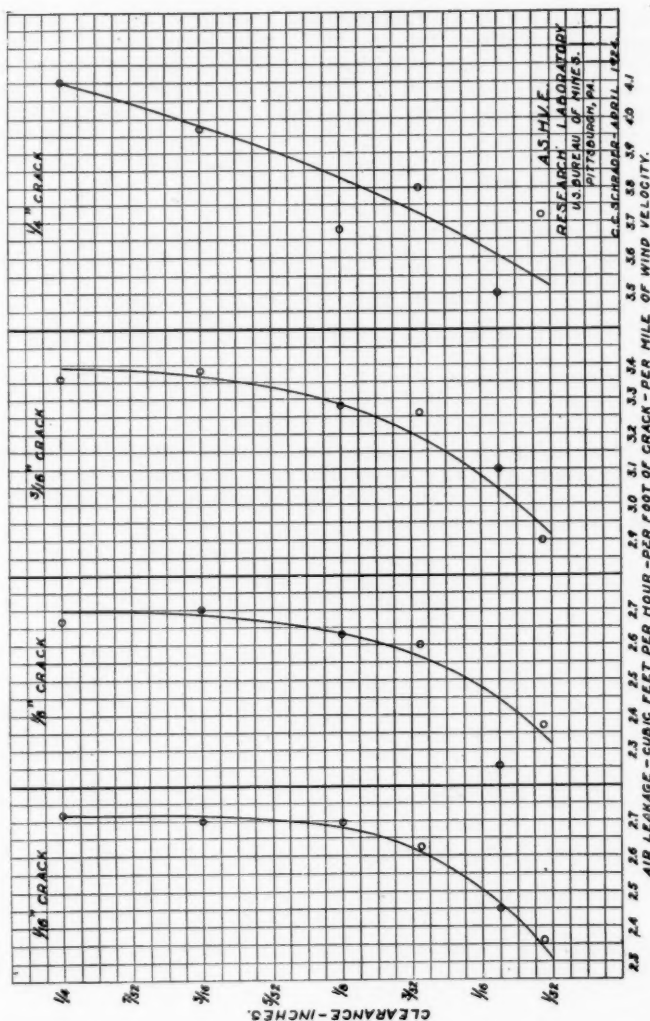


FIG. 7. LEAKAGE PER FOOT WITH VARIOUS CRACKS AND CLEARANCES. INTERLOCKING TYPE WEATHERSTRIP

With the groove $\frac{1}{32}$ in. wide a variation of 0.004 in. in width is given from one extreme to the other. Though most sashes at the present time are made of some less expensive wood, such as cypress or spruce, it is probable that the expansion of either would not be much greater than white pine. Therefore the clearance allowed here would be ample in any case.

No tables of leakage are included in this report because any values desired can



measuring and then averaging the clearances of a sufficient number of windows in buildings which are at least five years old. Different results would probably be obtained for sashes of different thicknesses because, although most windows leave the mill with the same clearance, those made of heavier wood will shrink more. Data obtained from twenty large manufacturers of windows have shown that almost all sashes from $1\frac{1}{4}$ in. to $2\frac{1}{2}$ in. thick are allowed $\frac{1}{16}$ in. clearance. It remains to be decided what the final clearance will be when the sashes have thoroughly weathered and dried.

The importance of the increase in clearance is shown by the following example: The capacity of a room $14 \times 12 \times 10$ ft. is 1680 cu. ft. With two windows such as those tested there would be 36.67 ft. of crack. From Fig. 5 the leakage for $\frac{1}{16}$ in. clearance is 6.6 c.f.m. per ft. of crack per mile of wind velocity. Assuming a 15 mile wind, the infiltration would be $6.6 \times 36.67 \times 15$ or 3630 cu. ft. per hr., or $\frac{3630}{1680} = 2.16$ air changes per hr. If the clearance increases to $\frac{1}{8}$ in. the leakage increases to 11.2 cu. ft. per hr. per ft. of crack per mile of wind velocity. The infiltration would then be $11.2 \times 36.67 \times 15$ or 6160 cu. ft. per hr., and the number of air changes would be increased to 3.66 per hr.

It must be remembered that in all these tests the crack between the frame and the setting was sealed, and this practice being easily done has become more and more common, and it has prevented considerable leakage, and thereby effected a saving in radiation surface and fuel.

FLOW OF STEAM AND CONDENSATION AS AFFECTED BY HIGH PRESSURES, HORIZONTAL OFFSETS AND VALVES

By LOUIS EBIN¹ AND R. L. LINCOLN,² PITTSBURGH, PA.

NON-MEMBERS

DATA contained in this article represent what has been gathered on Critical Velocities of Steam and Condensate Mixtures, since the last paper presented at the Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in New York City, January 1924. It covers three main factors of the general program:

- A. Effect of high pressure on maximum velocity.
- B. Effect of valves on maximum capacity.
- C. Effect of horizontal offsets on maximum velocity.

Effect of High Pressures on Maximum Velocity

The work upon flow of steam in pipes with counterflowing condensate thus far conducted and reported (JOURNAL A. S. H. & V. E., September 1922, January 1923, and February 1924), has all been done at practically atmospheric pressure. The actual range of pressures used was from zero to 4.0 in. water. In the greater number of cases all the required information was found before a pressure of 1 in. had been reached. Stating this a slightly different way the actual *pressure difference* existing in the system, at the point of maximum velocity, was found to be less than 1 in. of water. During all these tests the back of the radiator was open to the atmosphere and was therefore under atmospheric pressure. The capacity of the radiator was always much greater than the capacity of the pipe being tested, no steam escaping from the open radiator.

By using this method, that is, by conducting all tests at exceedingly low pressures and keeping the radiator open to the atmosphere, a very close regulation of the pressures and pressure differences could be maintained. This was very advantageous in obtaining characteristic curves of the relation of both velocity and capacity to the pressure and pressure differences existing in the system. This method was also of great aid in determining the exact point on the pressure velocity curves at which maximum velocity occurred and in noting the phenomena and effects taking place in the pipe throughout the range of pressure differences.

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Pressures as low as those used in our tests are not practical in actual installations; the pressures used in practice may go as high as 10 lb. gage. It was therefore decided to run a series of tests at higher pressures to note if possible the effect of variation in pressure upon the maximum velocity. Tests were run at pressures varying from slightly above atmospheric to 3 lb. inclusive, which was the limit of the equipment.

The equipment used in these tests was practically the same as that described in detail in the previous reports. There is, however, one difference between the tests conducted at atmospheric pressure, and those at higher pressures, that should be noted. Instead of keeping the radiator open to the atmosphere, an air valve was installed at the back of the radiator about one-third of the distance from the top as indicated in Fig. 1.

Flow of steam in a system is proportional, not to the header or boiler pressure, but to the pressure difference existing in the system. As has been previously

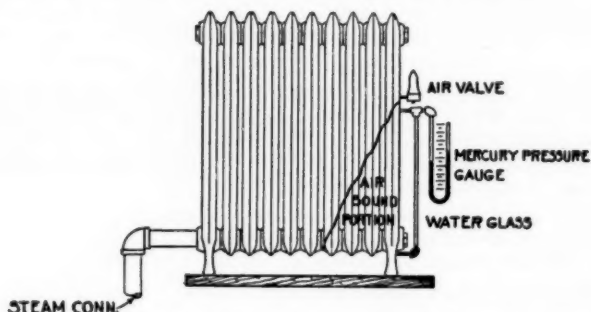


FIG. 1. DIAGRAM OF RADIATOR CONNECTIONS FOR TESTS USING HIGH PRESSURE STEAM

shown the pressure difference, at which the maximum velocity is obtained, is very small. In a system closed to the atmosphere the maximum velocity is determined by the pressure difference existing in that system. Furthermore in a system closed to the atmosphere, the pressure difference is fixed, depending upon the size of the radiator and pipe. For example, the maximum velocity for a 1 in. vertical pipe is obtained at a pressure difference of approximately 0.5 in. of water. If the size of the radiator is such that the pressure difference is smaller, the maximum velocity of the system will be smaller, or in other words, the pipe will not be able to deliver its maximum capacity. If the size of the radiator is increased so that a much greater pressure difference exists than that at which the maximum velocity is obtained, intermittent flow, water hammering, surging and so forth will result. Thus, to obtain the most effective, smooth and uniform operation, there must be such a relation between the size of the pipe and the size of the radiator, that the proper pressure difference will exist in the system.

In the tests upon the effect of high pressures it was necessary to vary the condensing power of the radiator from below the maximum to slightly above. Several methods were used to accomplish this. Water was sprayed over the radiator and the amount increased or decreased as required. A fan was used part of the time and the amount of air blown over the radiator was varied. A water glass was

connected at the back of the radiator, so that the amount of water collected in the radiator could be measured at any time. The pressure in the radiator was measured by a mercury U-tube also connected at the back as shown in Fig. 1.

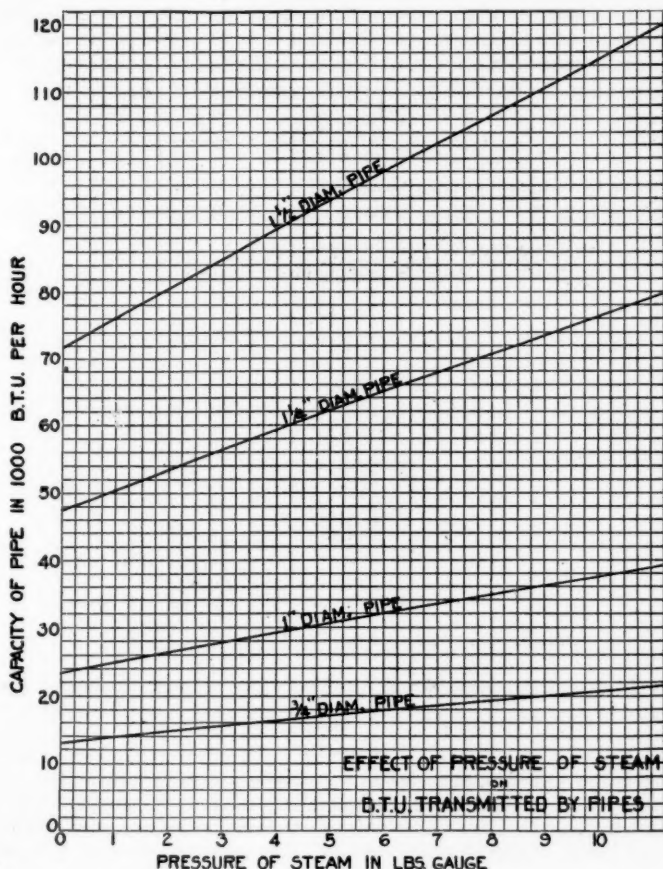


FIG. 2. VARIATION IN CAPACITY OF A PIPE WITH VARIATION IN PRESSURE

The results of the tests are given in Table 1 and the chief point of importance is the comparison between the maximum velocities obtained for the high pressure tests with closed radiators and the tests with very low pressures and open radiators. These comparisons are shown in the last three columns of the table. From the last column it may be noted that the maximum variation between the two conditions in all the tests was 10.1 per cent. When it is considered that this variation is but 1.6 ft. per second, and that the velocity varies greatly with a slight change

of pitch at the horizontal conditions, it may be seen that this variation is within the limits of accuracy of the conditions tested. Furthermore if the pressures in the header are compared with those in the radiator, it is noted that there are no differences indicated. The small pressure differences existing could not be measured by the mercury U-gage used. These results show that the maximum velocity is independent of the pressure in the system but is rather proportional to the pressure difference.

The velocity of steam in a pipe may be expressed by the equation:

$$V = \frac{W \times 12 \times 12}{60 \times 60 \times d \times A} = \frac{W}{25 Ad} \quad (1)$$

where V = velocity of steam in feet per second.

W = weight of condensate in pounds per hour.

A = inside area of pipe in square inches.

d = density of steam in pounds per cubic foot.

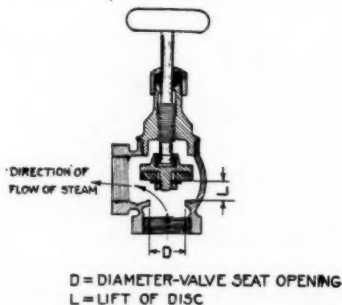


FIG. 3. DIAGRAM OF DISC TYPE OF ANGLE VALVES

The density of the steam d is a function of the pressure, (assuming dry and saturated steam) and increases with the increase of pressure. Thus from (1) as the pressure increases, if W is constant V must decrease, or if V does not change W must increase. To determine which of the two is constant and which changes, tests were run on a 1 in. diameter pipe at pressures varying from atmospheric to 3 lb. gage. The position of the pipe was kept constant. The results of the tests are given in Table 2. From this table it may be seen that the maximum condensate increases slowly as the pressure increases. The maximum velocity however remains practically constant, the greatest variation being $\frac{36.15 - 34.30}{36.15}$ or 5.1 per cent.

Since the actual pounds of steam carried by a pipe will increase as the pressure increases it is of interest to know the capacity of a pipe in B.t.u. as the pressure varies. At the same time that the density of the steam increases with the pressure the latent heat of vaporization decreases. However it decreases at a much slower rate than the density increases. The result is that the capacity of a pipe in B.t.u. will increase materially with the pressure. This is shown graphically by the curves in Fig. 2, and also by Table 3. For example, from Table 3 the increase in density

of the steam from atmospheric pressure to 10 lb. gage is 62 per cent, the decrease in latent heat is only 2 per cent. The result is a gain of 60 per cent in the B.t.u. transmitted at a pressure of 10 lb. over that transmitted at atmospheric pressure.

There is one other point that was noted in connection with the high pressure tests that may be of interest. In a number of the tests that were conducted with

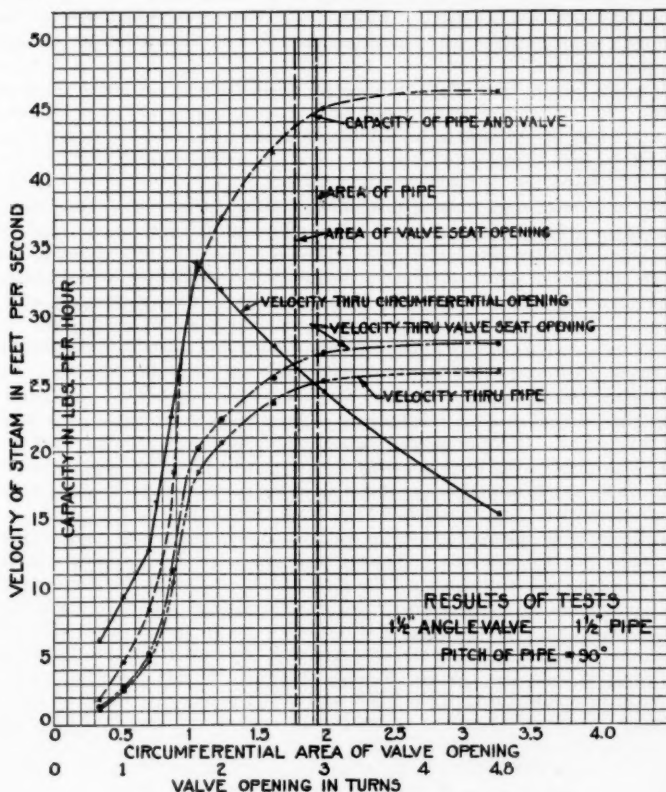


FIG. 4. RESULTS OF TESTS ON ANGLE VALVES

the air valve on the radiator it was noted the region directly below the air valve and for some distance back, Fig. 1, was cold. In other words, air being heavier than steam, this region was simply air bound. From this it would seem that the efficiency of a radiator might be improved if the air valve was placed as low as is consistent with its safe working.

Effect of Valves upon Maximum Capacity

A series of tests was run on valves of various types and sizes to determine the factors that affect the flow of steam through a valve. Tests were run on (1) angle valves, (2) globe valves and (3) gate valves, in sizes from $\frac{3}{4}$ to $1\frac{1}{2}$ in. inclusive.

The results show some interesting facts and some points that should be of assistance in designing and using valves for single pipe work. All the valves tested were bought in the open market and include several different makes.

Angle Valves. Fig. 3 shows a diagram of the disc type of angle valve as used in the tests. This type of valve is very extensively used where it is necessary to place a shut-off valve at a right angle turn in the steam line. Perhaps the commonest type is the angle radiator valve. The tests on the angle valves were all made with the pipe in a vertical position and the valve connected to the radiator by a short nipple.

The main factors that affect the flow of steam through an angle valve are as follows:

1. Lift of the disc or circumferential area.
2. Area of seat opening.
3. Resistance of valve or valve friction.

Effect of circumferential area on maximum velocity. Fig. 4 shows the results of the tests upon the $1\frac{1}{2}$ in. pipe and valve. The velocities through various parts of the valve and pipe are plotted against the circumferential area in square inches. The circumferential area is a direct function of the lift of the disc, being the product of the lift of the disc and the circumference of the valve seat opening, and thus the curves will vary in exactly the same manner for the lifts of the disc. These curves are typical in form for angle valves of all sizes tested and will therefore serve as a study of the velocities through angle valves in general.

As the valve is first opened one turn, for example, the area of the circumferential opening is so small and the resistance is so great that the capacity in pounds per hour is very low. In other words this portion of the valve opening is very ineffective. This will be even more apparent for valves that have the loose or removable disc. These valves always have some play in the movement of the disc decreasing the effective area very considerably at small openings.

As the valve is opened further the velocity and capacity will increase rapidly, practically in a straight line or directly proportional to the circumferential area. This procedure will continue until the ratio of the capacity to the circumferential area is maximum, at which point the velocity through the valve circumference will become maximum. As the valve is still further opened the capacity will continue to increase but at a reduced rate, and since the circumferential area is increasing at a constant rate the velocity through the valve circumference will decrease. The velocity curves through the valve seat opening and pipe will tend to flatten out until a point is reached at which the circumferential area is slightly greater than the valve seat area. At this point the capacity of the pipe and valve, and the velocity through the valve seat opening and pipe become maximum. This condition was found true for all tests on all sizes. As soon as the circumferential area of the valve became slightly greater than the area of the valve seat opening, the capacity of the system became maximum. Beyond this point any further increase in circumferential area did not affect the capacity of the pipe and valve.

Effect of area of valve seat opening on maximum velocity. In all cases the area of the valve seat opening of commercial valves was found to be smaller than the area of the correspondingly sized pipe. The area of the valve seat opening varied in different sizes and kinds of valves from 98.4 per cent to 82.2 per cent of the area of the pipe. There was no definite relation found between the area of the valve seat opening and the area of the pipe for different sizes of valves. Consequently there is no definite relation between the capacity of a pipe when tested alone, and the capacity of a pipe and valve for different sizes.

The results of the tests on angle valves are given in Table 4, and are given for a maximum lift of the disc. It will be noted that the capacity of the pipe and valve is very much lower than the capacity of the pipe alone. The two factors that might explain this are (1) decrease in area of valve seat over area of pipe, and (2) valve resistance or friction. But while the area of the valve seat opening varies from 98.4 per cent to 82.2 per cent of the area of the pipe, the capacity of the pipe and valve varies from 95.0 per cent to 63.1 per cent of the capacity of the pipe alone. The difference between the two may be attributed to valve resistance.

A series of tests was run to determine the variation in the capacity of a valve with the variation in valve seat area. A test was run with the valve seat opening a certain area and the maximum capacity determined. The area of the valve seat opening was then increased and the test repeated. The results of these tests are shown in Fig. 5. It may be seen that the capacity of the valve increases directly as the area, the curves flattening out as the area of the valve seat opening approaches the area of the pipe. If the area of the valve seat opening is increased still further, the capacity instead of remaining constant will decrease slightly.

The work on Hydraulic Experiments with Valves by the University of Illinois has brought out the point that there is a definite relation between the lift of the disc and the area of the valve at which the resistance is a minimum. Either above or below this point the resistance will increase. In our tests the energy in the steam to cause flow is maximum when the area of the valve seat opening is equal to the pipe area. As the area of the valve seat opening is increased a portion of this energy is required to overcome the additional resistance of the valve. Thus beyond this point the capacity will decrease slightly.

It is interesting to note that if the valve used is one size larger than that of the pipe, the capacity of this valve and pipe will be practically equal to the capacity of the pipe alone. Angle valves of sizes 1, $1\frac{1}{4}$, and $1\frac{1}{2}$ in. were tested to determine this, each with the pipe one size smaller. In all cases the lift of the disc was maximum. The results of the tests are given in Table 5.

The area of the valve seat opening is much greater than the area of the pipe. The circumferential area is also much greater. It therefore seems probable that the resistance through the valve has become negligible in comparison with the pipe resistance. The limiting factor is the area of the pipe, and consequently the capacity of the valve and pipe has become equal to the capacity of the pipe alone.

Valve Resistance. Due to the large number of variables that enter into the flow of steam through a valve it is exceedingly difficult to separate the effects due to valve resistance. It has therefore been found advisable to treat the subject of valve resistance in combination with the other factors. Wherever it is possible to separate it or to point out its significance it will be done.

Globe Valves. Fig. 6 shows a diagram of a typical globe valve as used in our tests. Four sizes of this type of valve were tested, $\frac{3}{4}$ in. to $1\frac{1}{2}$ in. inclusive, each in five different positions:—

- | | |
|--|-------------------------|
| (1 and 2) pitch of pipe approximately 0 deg. | { valve stem vertical |
| | { valve stem horizontal |
| (3 and 4) pitch of pipe approximately 0.40 in. per ft. | { valve stem vertical |
| | { valve stem horizontal |
| (5) pitch of pipe approximately 90 deg. | |

There is considerable difference between the capacity of a globe valve when the valve stem is vertical, and when the valve stem is horizontal as will be shown later.

The usual method of placing a globe valve in a line is such that the line pressure will be on the under side of the disc when the valve is closed. All the valves were tested in this position. The direction of the flow of steam is indicated by the arrows in the diagram. During all the tests the valve was wide open, that is the lift of the disc was maximum.

The results of all the tests are given in Table 6. The capacity of the valve in pounds per hour, and the velocity of the steam through the valve seat opening in feet per second are given for each size of the valve, and for each position tested. The ratio of the capacity of the pipe and valve to the capacity of the pipe alone is

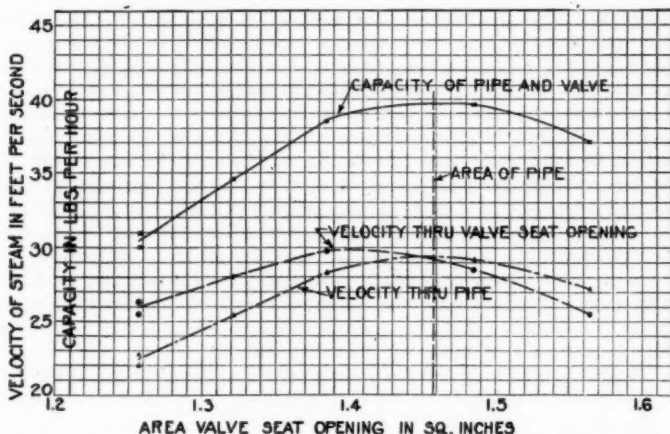


FIG. 5. VARIATION IN CAPACITY OF AN ANGLE VALVE WITH VARIATION IN VALVE SEAT AREA

also given. This ratio is a measure of the efficiency of the valve. The results of the tests may be considered under the following divisions:

1. Effect of the area of the valve seat opening upon the maximum capacity.

On Fig. 7 the capacities and velocities of the valves tested are plotted against the area of the valve seat opening. This area being smaller than either the area of the pipe or the circumferential area of the valve is the most important factor in determining the capacity of the valve. One separate curve is given for each position of the pipes tested. In the tests, when the pipe was in an approximately horizontal position, all conditions not being exactly at 0 deg., it was necessary to correct the results and bring them all to the same pitch (0°) before they could be compared.

It is seen from the curves that for the same position of the pipe the capacity varies directly as the area of the valve seat opening. The velocity, however, is independent of the valve seat area and within certain limits is practically constant at each pitch for all sizes tested. The greatest variation is found for the $\frac{3}{4}$ valve at the horizontal pitches of the pipe. This is due in a large measure, first to the difficulty of obtaining an absolutely true 0 deg. pitch, second to the great effect of any small variations in pitch, and third to the very small total capacities obtained at the horizontal pitches.

2. Effect of position of valve stem on maximum capacity.

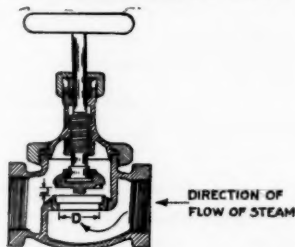
It is common knowledge and practice that a globe valve should not be set in a horizontal line with the valve stem vertical. This fact is emphasized experimentally by comparing from Table 6 the results of the tests on the two conditions, valve stem vertical, and valve stem horizontal. When the valve stem is vertical the capacity of the pipe and valve is reduced to approximately $\frac{1}{4}$ of the capacity of the pipe alone.

This reduction in capacity varies from 31.2 per cent to 35.7 per cent and appears to be independent of size of valve or pitch of pipe. The exception to the case is the $\frac{1}{4}$ in. valve. However, it has been pointed out that, at very small pitches, the results of the tests for $\frac{3}{4}$ in. pipe and valve are much more difficult to obtain and are not as consistent as for the larger size pipes and valves.

When the valve stem is horizontal, the per cent of the capacity of the pipe varies from 57.6 to 88.3. It now varies with the size of the valve and perhaps to some extent with the pitch of the pipe. However, the smallest capacity, with the valve stem horizontal, 57.6 per cent is practically twice as large as the capacity with the valve stem vertical. In general, placing a globe valve in a horizontal line with the valve stem vertical will reduce the capacity of the line to 40 to 60 per cent of the capacity with the valve stem horizontal.

3. Effect of size of valves upon maximum capacity.

Comparing from Table 6 the efficiencies of the globe valves, that is the ratios of the capacities of the valves to the capacities of the pipes, it is noted that the values run highest for the $\frac{3}{4}$ in. valve, decreasing slightly for the 1 in. size. However, for the



D=DIAMETER-VALVE SEAT OPENING
L=LIFT OF DISC

FIG. 6. DIAGRAM OF TYPICAL GLOBE VALVE

larger sizes of valves ($1\frac{1}{4}$ in. and $1\frac{1}{2}$ in.) the drop in the efficiency is much greater, which was entirely unexpected, and seemed to indicate that the resistance through a valve was greater for the larger sizes. The only explanation that may be offered is that the smaller sizes of valves are better proportioned for the flow of steam and condensate with as small a loss due to resistance as possible.

4. Effect of proportions.

Very little data has come to the attention of the Laboratory on the various factors that enter into the loss of head through a valve. There can be little doubt but that the proportions of the valve, form, shape of passageways, ratios of lift of disc to valve areas and other factors all are very important in determining this loss.

The results here indicate that for one-pipe work the proportions of the valve are more ideally approached, in the smaller sizes of valves ($\frac{3}{4}$ in. and 1 in.) than in the larger sizes ($1\frac{1}{4}$ in. and $1\frac{1}{2}$ in.). This can only be proven conclusively by an exhaustive set of experiments involving all the factors that go to make a valve.

An interesting comparison as to the variation in the capacities and efficiencies of valves with the variation in proportions is shown in the tests on two valves of the same nominal size but of different makes. Both valves were tested wide open and with the pipe in a vertical position. The results are given below:

	Size Inches	Area Valve Seat	Circum. Pipe	Area Pipe	Cond. Lb./Hr. Valve Alone	% Effi- ciency	Maximum Velocity—		
		Sq. In.		Sq. In.			Valve	Valve Seat	Circum. Pipe
Valve A...	$\frac{3}{4}$	0.4418	0.743	0.5373	12.00	88.6	29.10	17.2	23.9
Valve B...	$\frac{3}{4}$	0.4418	0.357	0.5373	9.30	68.7	22.60	27.9	18.52

The difference between the efficiencies of the two valves is 88.6-68.7 or 19.9 per cent. In other words valve *A* will permit 20 per cent more of the capacity of the pipe to pass through than will valve *B*. The area of the valve seat is the same for both valves. The circumferential area or lift of disc is, however, much greater for valve *A* than *B*. Furthermore the body of valve *A* is larger and with freer

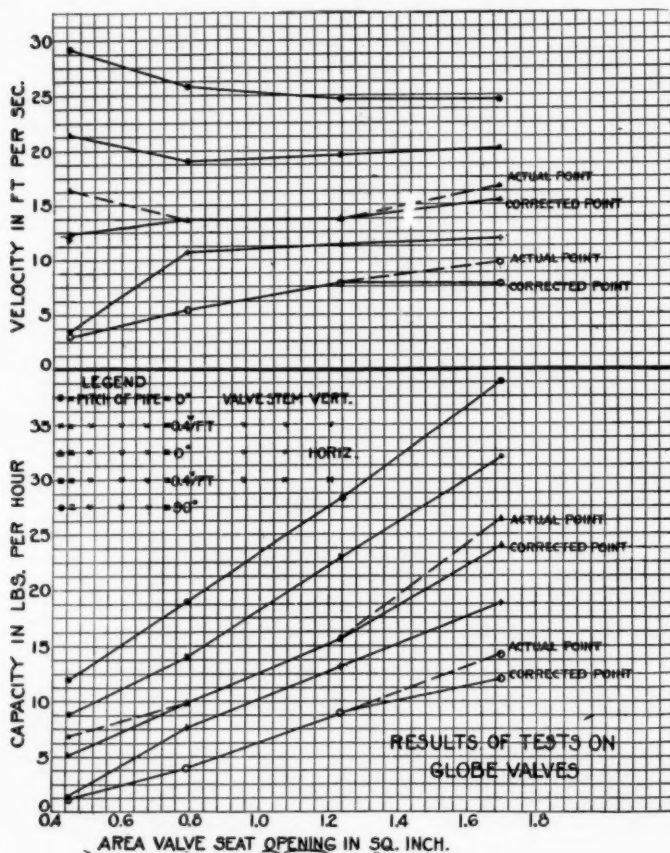


FIG. 7. RESULTS OF TESTS ON GLOBE VALVES

passages than that of valve *B*. The decreased capacity of valve *B* may therefore be attributed to its very small circumferential area and its greater resistance. This test will also serve to bring out the difficulties involved in obtaining reliable standards and comparable data.

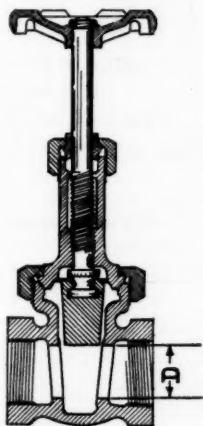
Gate Valves. The gate or straightway valve is the most commonly used of all valves. In its simplest form a wedge or vertical disc moves up and down in the body of the valve either opening or closing it. A diagram of a typical form of

such a valve is shown in Fig. 8. It may be seen from the diagram that the travel of the steam, or steam and condensate has no such changes in direction as found either in the angle or globe valves. Consequently the resistance through this type of valve should be much smaller than for either of the other types.

Four sizes of gate valves were tested $\frac{3}{4}$ in. to $1\frac{1}{2}$ in., and each valve was tested at three different pitches:

1. Pitch of pipe approximately 0 deg.
2. Pitch of pipe approximately 0.4 in. per ft.
3. Pitch of pipe approximately 90 deg.

From the nature of the construction of the valve the position of the valve stem has no effect. As was found for both angle and globe valves the areas of the open-



**D = DIAMETER OF VALVE OPENING
= SMALLER THAN DIAMETER OF
CORRESPONDINGLY SIZED PIPE**

FIG. 8. DIAGRAM OF TYPICAL GATE VALVE OR STRAIGHTWAY VALVE

ings through the valves are smaller than the areas of the corresponding pipes, and in about the same ratios. The results of the tests are given in Table 7.

There are two main factors which affect the flow of steam through a gate valve (1) area of valve opening, and (2) resistance of valve. It has been shown in previous reports that for horizontal pipes and especially pipes of small pitches the maximum capacity is greatly affected by the pitch, and by the length of the pipe. Both of these factors are more pronounced in their effect than the area. In other words any small variation in area is negligible in its effect on the capacity. Thus it may be seen from Table 7 that, with one or two exceptions, the capacities of the valves at the small pitches are practically equal to the capacities of the pipes.

When the pitch of the pipe, however, is 90 deg. the area has become the main factor, and from the table it may be noted that the capacity of the valve and pipe varies from 68.6 per cent to 78 per cent of the capacity of the pipe. This variation

is partly due to the decrease in the area of the valve over the area of the pipe, and partly to the resistance of the valve. If the capacity of the pipe and valve is compared to the capacity of a pipe alone whose area is equal to the area of the valve, it is seen that the capacity of the valve is approximately 87 per cent of the capacity of this pipe. The $\frac{3}{4}$ in. valve is the one exception. Thus, for example the capacity of a 1 in. valve is 18.78 lb. per hr. The capacity of a pipe whose area is equal to the area of a 1 in. valve is 21.5 lb. The valve will then transmit 87.5

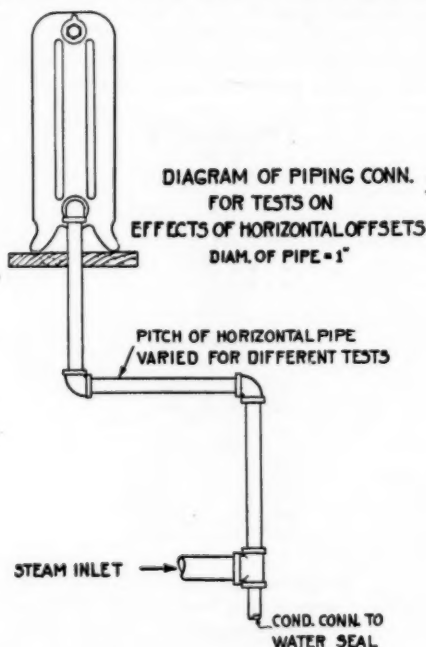


FIG. 9. DIAGRAM OF PIPING CONNECTIONS FOR TESTS ON HORIZONTAL OFFSETS

per cent of the capacity of a corresponding pipe. The remaining 12.5 per cent must be attributed to valve friction.

General Discussion of Valves

Table 8 shows a comparison of the three types of valves tested and includes the capacity of each valve tested, with the percentage of the capacity of the valve compared with the capacity of the pipe. All the results are for a vertical position of the pipe. The table indicates that for the two smaller sizes of valves there is very little difference in efficiency of the three types of valves. For these sizes the results for the angle and globe valves run much higher than was anticipated due to the better proportioning of the valves. For the two larger sizes the results of the globe and angle valves have dropped materially below those of the gate valves.

The general average is higher for the gate valves than for the angle valves and lower for the globe valves.

The variation in efficiencies with the size is very pronounced for the globe and angle valves but almost negligible for the gate valves.

TABLE 1. RESULTS OF TESTS USING HIGH PRESSURE STEAM

Nominal size of pipe, in.	Pitch of pipe in degrees	Header pressure lb./sq. in.	Radiator pressure lb./sq. in.	Maximum condensate lb./hr.	Maximum velocity ft./sec.	Maximum velocity with radiator open to atmosphere	Per cent change from maximum*
1	90	0.25	0.25—	24.49	30.1	30.1	00.0
1	90	0.50	0.50—	26.35	31.8	30.1	+ 5.8
1 ¹ / ₄	90	0.50	0.50—	49.90	34.2	35.2	- 2.8
1 ¹ / ₂	90	1.00	1.00—	72.58	36.4	38.5	- 5.4
1 ¹ / ₂	- 0.05	1.00	1.00—	27.27	13.7	14.6	- 6.2
1 ¹ / ₄	0.0833	1.00	1.00—	21.30	14.2	15.8	-10.1
1	18.0833	1.00	1.00—	31.63	37.2	36.5	+ 1.9

* Note: Per cent change from maximum equals velocity of pipe with closed radiator minus velocity with radiator open to atmosphere, divided by velocity with radiator open to atmosphere.

The valve resistance is greatest for the globe, somewhat less for the angle valve, and very much less for the gate valve.

It is a difficult matter to draw comparisons between results where there are so many variables of such different kinds. Perhaps the most important point to be indicated by the tests is the need for much more research and a more detailed study of valves. There seems to be considerable that is not yet understood in the design of valves for best efficiency.

Effect of Horizontal Offsets on the Maximum Velocity

It is frequently found necessary in heating practice to offset a riser because of obstructions, or frequently required to run a long horizontal connection to a radiator. What would be the effect of such conditions on the maximum velocity? This problem really resolves itself into the effect on the maximum velocity of a combined system of horizontal and vertical pipes. It may be solved from the data on the flow of steam in pipes of various pitches previously published.

TABLE 2. EFFECT OF PRESSURE ON MAXIMUM VELOCITY

1" steel pipe, 5'0" long—inside diameter = 1.046"

Pitch of Pipe = 18.08 deg.

Pressure above atmospheric lb./sq. in.	Density of steam lb./cu. ft.	Maximum condensate lb./hour	Maximum velocity feet/second
0.00	0.0373	29.00	36.15
0.10	0.0375	28.59	35.45
0.25	0.0379	29.25	35.90
0.50	0.0388	29.77	35.99
1.00	0.0396	30.25	35.60
2.00	0.0420	31.15	34.60
3.00	0.0443	32.58	34.30

To determine the effect experimentally, three pieces of 1 in. pipe were connected together and the combined system tested. Fig. 9 shows a diagram of the position of the pipes as tested. The system consists of two vertical lines with a horizontal connection. The pitch of the horizontal pipe was varied for different tests.

The results of the test are given in Table 9. The table shows the pitch of the

horizontal line as tested, the maximum capacity in pounds per hour, and the maximum velocity in feet per second for the combined system. It also shows the maximum velocities for both the vertical and horizontal pipes when tested separately.

It has been demonstrated in several instances in previous reports that the capacity of a system is limited by the capacity of the smallest portion of that system. Thus, for example, if the capacity of one part of a system taken singly is 30 lb. per hr., while that of another part is but 15 lb. per hr., the capacity of the combined system will be equal to 15 lb. per hr.

If the maximum velocity for the combined pipes is compared with that of either

TABLE 3. VARIATION IN CAPACITY OF A PIPE IN B.T.U. PER HOUR WITH THE VARIATION IN PRESSURE
Based on Standard Inside Diameter of Pipes
Pitch = 90 deg.

Pressure steam above atmos. lb./sq. in.	Density steam lb./cu. ft.	Latent heat of vapor- ization B.t.u./lb.	Maximum capacity				Maximum capacity			
			$\frac{1}{4}$ "	1"	$\frac{1}{4}$ "	$\frac{1}{2}$ "	$\frac{1}{4}$ "	1"	$\frac{1}{4}$ "	$\frac{1}{2}$ "
0 +	0.0373	970.4	13.45	24.25	49.15	74.03	13,050	23,550	47,650	71,900
0.25	0.0379	969.8	13.68	24.65	50.00	75.3	13,280	23,900	48,500	73,000
0.5	0.0388	969.3	13.87	25.00	50.70	76.5	13,460	24,200	49,150	74,100
1.0	0.0396	968.2	14.29	25.75	52.20	78.65	13,840	24,950	50,600	76,150
2.0	0.0420	966.2	15.12	27.25	55.20	83.30	14,610	26,350	53,400	80,500
3.0	0.0443	964.3	15.97	28.80	58.40	87.90	15,400	27,780	56,300	84,600
4.0	0.0467	962.4	16.84	30.35	61.50	92.70	16,210	29,200	59,200	89,300
5.0	0.0491	960.5	17.70	31.88	64.65	97.50	17,000	30,650	62,150	93,600
8.0	0.0561	955.6	20.20	36.42	73.80	111.30	19,300	34,800	70,500	106,500
10.0	0.0606	952.4	21.85	39.40	79.90	120.4	20,800	37,550	76,100	114,900

the vertical or the horizontal pipe, it will be found in each case that the maximum velocity for the combined pipes will be practically equal to the smaller of the other two values. For example, when the pitch of the horizontal line is 0 deg., the velocity in the horizontal line is much smaller than the velocity in the vertical line, and it will be the limiting factor. Thus the velocity of the combined system was found to be 17.8 ft. per second, and the velocity of the horizontal pipe alone 16.5 ft. per second. The velocity of the vertical pipe alone was 29.8 ft. per second.

Again when the system was tested with the horizontal pipe at a pitch of 45 deg., the velocity through the horizontal pipe was 46.8 ft. as compared with 29.8 ft. or the vertical pipe. In this test the vertical pipe became the limiting factor and the velocity for the combined system was found to be 30.2 ft. per second.

The most important points of this report may be summarized as follows:

Effect of High Pressures

1. The maximum velocity of the steam in a pipe is independent of the pressure existing in the system.
2. The capacity of a pipe in B.t.u. will increase with the pressure. Thus, for example, a pipe will transmit about 30 per cent more B.t.u. at 5-lb. gage pressure than at atmospheric pressure.

Effect of Valves

1. The capacity of a valve will vary with the size of the valve opening, lift of disc, and type of valve.
2. The area of the valve seat opening in all valves tested was found to be smaller than the area of the correspondingly sized pipe. Consequently the capacity of a valve was in all cases found to be smaller than the capacity of the pipe.

TABLE 4. RESULTS OF TESTS ON ANGLE VALVES

Nominal pipe size inches	Area of pipe sq. in.	Area valve seat opening sq. in.	Per cent of area of pipe	Maximum Capacity Pipe and valve	Capacity Pipe alone	Per cent of capacity of pipe alone
$\frac{3}{4}$	0.537	0.4418	82.2	9.68	13.52	71.6
1	0.835	0.822	98.4	22.10	23.30	95.0
$1\frac{1}{4}$	1.459	1.258	86.2	30.00	47.5	63.1
$1\frac{1}{2}$	1.927	1.773	92.2	46.13	68.5	67.4

TABLE 5. RESULTS OF TESTS ON ANGLE VALVES WITH VALVE ONE SIZE LARGER THAN SIZE OF PIPE

Pitch of Pipe = 90 deg.

Nominal Size Pipe	Valve	Area in Sq. In. Pipe	Valve seat opening	Capacity in lb./hr. Pipe alone	Pipe and valve	Per cent of capacity of pipe alone
$\frac{3}{4}$	1	0.5373	0.822	13.52	13.33	98.6
1	$1\frac{1}{4}$	0.835	1.258	23.30	24.58	100.0
$1\frac{1}{4}$	$1\frac{1}{2}$	1.459	1.773	47.50	46.38	97.5

TABLE 6. RESULTS OF TESTS ON GLOBE VALVES

Pitch = 0 deg.

Nominal size	Valve Stem Vertical			Valve Stem Horizontal		
	Capacity of pipe and valve lb./hr.	Per cent of capacity of pipe alone	Vel. thru valve seat opening ft./sec.	Capacity of pipe and valve lb./hr.	Per cent of capacity of pipe alone	Vel. thru valve seat opening ft./sec.
$\frac{3}{4}$	1.19	18.00	2.9	6.80	88.1	16.5
1	3.96	32.95	5.4	9.88	82.2	13.5
$1\frac{1}{4}$	9.00	34.35	7.8	15.70	60.0	13.6
$1\frac{1}{2}$	14.38	33.15	9.1	26.47	61.6	16.7
$\frac{3}{4}$	1.38	11.8	3.4	8.83	76.3	21.5
1	7.73	35.7	10.6	14.06	66.1	19.2
$1\frac{1}{4}$	13.20	31.2	11.4	22.88	54.3	19.9
$1\frac{1}{2}$	18.81	33.0	11.9	32.00	56.7	20.2
						12.00
						88.7
						79.0
						29.1
						25.9
						24.7
						24.6

TABLE 7. RESULTS OF TESTS ON GATE VALVES

Nominal size of pipe and valve inches	Area of pipe sq. in.	Area of valve sq. in.	Pitch = 0°		Pitch = 0.4 in./ft.		Pitch = 90°		Column A*
			Capacity of pipe and valve lb./hr.	Per cent of capacity of pipe alone	Capacity of pipe and valve lb./hr.	Per cent of capacity of pipe alone	Capacity of pipe and valve lb./hr.	Per cent of capacity of pipe alone	
$\frac{3}{4}$	0.5373	0.4418	5.25	79.4	10.15	85.9	10.55	78.0	97.5
1	0.860	0.809	12.71	100	22.50	94.0	18.78	78.0	87.5
$1\frac{1}{4}$	1.459	1.208	28.50	100	33.13	100	32.50	68.6	87.1
$1\frac{1}{2}$	1.927	1.743	35.00	100	50.93	90.4	52.00	75.9	86.2

* Column "A" = Capacity of pipe and valve divided by capacity of a pipe whose area equals area of valve seat opening.

3. The maximum capacity of an angle valve was found at a point when the circumferential area of the valve was approximately equal to the area of the valve seat. Beyond this point any further lift of the disc had no effect.

4. If an angle valve is used one size larger than the size of the pipe, the capacity of the valve and pipe will be equal to the capacity of the pipe. This would be one method of obtaining the maximum capacity of a pipe.

5. A globe valve should never be set in a horizontal line with the valve stem in a vertical position. This will reduce the capacity of the valve from 40 per cent to 60 per cent.

6. Indications are that the smaller sizes of globe and angle valves ($\frac{3}{4}$ in. and 1 in.) are much more efficient than the large sizes ($1\frac{1}{4}$ in. and $1\frac{1}{2}$ in.).

TABLE 8. COMPARISON OF THE RESULTS OF TESTS ON THREE TYPES OF VALVES

Pitch of Pipe = 90 deg.

Nominal size inches	Angle Valve		Globe Valve		Gate Valve	
	Capacity lb./hr.	Per cent capacity of pipe alone	Capacity lb./hr.	Per cent capacity of pipe alone	Capacity lb./hr.	Per cent capacity of pipe alone
$\frac{3}{4}$	9.68	71.5	12.00	88.6	10.55	77.9
1	22.10	95.1	19.00	79.0	18.78	78.0
$1\frac{1}{4}$	30.00	63.8	28.44	60.0	32.50	68.6
$1\frac{1}{2}$	46.13	67.4	38.91	56.8	52.00	75.9

TABLE 9. RESULTS OF TESTS ON HORIZONTAL OFFSETS

Pitch of horizontal line	Maximum capacity lb./hr.	Maximum Velocity Feet per Second		
		Combined system	Vertical pipe alone	Horizontal pipe alone
0°	13.83	17.8	29.8	16.5
0°	13.28	17.1	29.8	16.5
2.475 in./ft.	20.94	26.9	29.8	25.4
2.89 in./ft.	21.25	27.3	29.8	26.1
4.10 in./ft.	19.53	25.2	29.8	27.1
45°	23.44	30.2	29.8	46.8

This is probably due to the better proportioning of the smaller sizes of valves.

7. In the smaller sizes of valves ($\frac{1}{4}$ in. and 1 in.) there is very little difference in efficiency among the gate, globe and angle valves. In the larger sizes ($1\frac{1}{4}$ in. and $1\frac{1}{2}$ in.) the efficiency of the gate will be higher than either the globe or angle valves.

8. The capacity of two different makes of the same size valve may vary as high as 20 per cent due to different proportions used in the two valves.

9. The capacities of valves in general may be increased, first by making the area of the valve seat at least equal to area of the pipe, and, second by a better study of the proportions of the valves so as to cut valve resistance to a minimum.

Effect of Horizontal Offsets.

The capacity of a combined system of horizontal and vertical pipes will be equal to the capacity of the smallest portion of the system. A horizontal offset should therefore be given sufficient pitch so that the capacity of the offset will be as close as possible as the capacity of the vertical pipe.

Acknowledgment

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No. 706

EFFECTIVE TEMPERATURE APPLIED TO INDUSTRIAL VENTILATION PROBLEMS*

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THE object of this paper is two-fold. In the first part it presents in a concise form the completed work on effective temperature with moving air as determined by subjects practically at rest, stripped to the waist, and facing the current of air; while the second part it is devoted to industrial applications of the experimental facts, for improving working conditions, and thereby increasing the output in various industries where adverse heat conditions constitute one of the most serious considerations.

A progress report³ on the work was presented at the Annual Meeting of the Society in January, 1924. The report covered in detail, apparatus, methods, and results from well below the comfort zone to body temperature. Since then the work has been extended to cold temperatures, as well as to temperatures considerably above that of the body, and this paper contains information from 30 to 170 deg. dry-bulb temperature, and supersedes the previous publication.

Test Conditions

It is not the purpose of this writing to go into preliminaries and details of the tests. For such information the reader is referred to the JOURNAL, for February, 1924. It is sufficient to say here that the experiments were conducted in two wind tunnels, built for the purpose in the psychrometric rooms of the Research Laboratory. Temperature conditions in one tunnel with still air were found equally warm or equally comfortable to other temperature conditions in the second tunnel with moving air, both of the same relative humidity but of different dry bulb temperature. A number of such equivalent conditions with constant velocity plotted on a psychrometric chart constitute a so-called *Effective Temperature Line*, which has the same numerical value as the one drawn through the corresponding equivalent conditions with still air.

* Work carried out in conjunction with the U. S. Bureau of Mines of its Pittsburgh Experiment Station.

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³ Cooling Effect on Human Beings Produced by Various Air Velocities, by F. C. Houghten and C. P. Yagloglou.

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General Characteristics and Relation of the Effective Temperature Lines to Dry- and Wet-Bulb Temperature

Three psychrometric charts with effective temperature lines from 30 to 115 deg. superimposed are shown in Figs. 1, 2, and 3 for velocities of 150, 300, and 500 ft. per min., respectively. As a distinguishing feature, the lines are drawn heavy and not extended down to the base line, but stop at the lowest humidity obtained in the tests, in the proximity of which is given their numerical values as fixed by those of the corresponding effective temperature lines for still air. The latter are shown by the straight lines of medium thickness in contrast to the wet-bulb lines which are the thinnest on the chart.

It will be observed that unlike the effective temperature lines with still air, the lines for moving air are not straight all over the chart, but are distinctly curved for temperatures from above the comfort zone to body temperature. At low temperatures the effect of the wind is to increase heat loss by evaporation and convection. The result is that the surface of the body becomes dry, and the body approaches the thermal properties of a dry-bulb thermometer, as shown by the effective temperature lines becoming straight and parallel to the dry-bulb lines. At high temperatures, when the surface of the body is completely wet with perspiration, the effect of the wind is to change the temperature of the surface so that it approaches the wet-bulb temperature of the air. The latter thus becomes the predominant factor of the severity of exposure, and the effective temperature lines flatten out and become parallel to the wet-bulb lines.

For intermediate temperatures the degree of wetness of the body surface varies from an appreciable state of perspiration at high to insensible perspiration at low humidities. This property of the human body causes the heat loss by evaporation to vary due to air motion, depending upon the amount of moisture available, as a result of which the lines curve downward with decreased humidities.

No marked change is observed in the curvature of the lines with increased velocities, but they become steeper as the velocity increases for reasons previously discussed.

Determining Effective Temperature

The effective temperature of a condition is determined by the line passing through the point of intersection of the dry-bulb and wet-bulb temperature on the chart corresponding to the velocity of the air. The difference between the effective temperature of the condition with still air and that with the given velocity, is a measure of the cooling produced by the particular movement. Similarly, the difference between the dry-bulb temperature of the condition and the dry bulb of the corresponding equivalent condition with still air gives the cooling in degrees dry-bulb temperature.

Assuming air at 80 deg. dry bulb and 71 deg. wet bulb moving at a velocity of 300 ft. per min., Fig. 2 shows that the effective temperature will be 67 deg.; the effective temperature with still air 74.7 deg.; the cooling produced by air motion 74.7 deg. - 67.0 deg. = 7.7 deg. effective temperature; the dry-bulb temperature of the equivalent condition with still air 71.0 deg.; and the cooling in degrees dry-bulb temperature 80.0 deg. - 71.0 deg. = 9.0 deg.

The cooling produced by any velocity varies with both dry- and wet-bulb temperatures. For ordinary temperatures it is maximum at saturation and minimum at low humidities, due to the fact that in the latter case the capacity of air for ab-



FIG. 1. PSYCHROMETRIC CHART SHOWING EFFECTIVE TEMPERATURE LINES WITH STILL AIR AND WITH 150 FT. PER MINUTE VELOCITY SUPERIMPOSED



FIG. 2. PSYCHROMETRIC CHART SHOWING EFFECTIVE TEMPERATURE LINES, WITH STILL AIR AND WITH 300 FT. PER MINUTE VELOCITY, SUPERIMPOSED

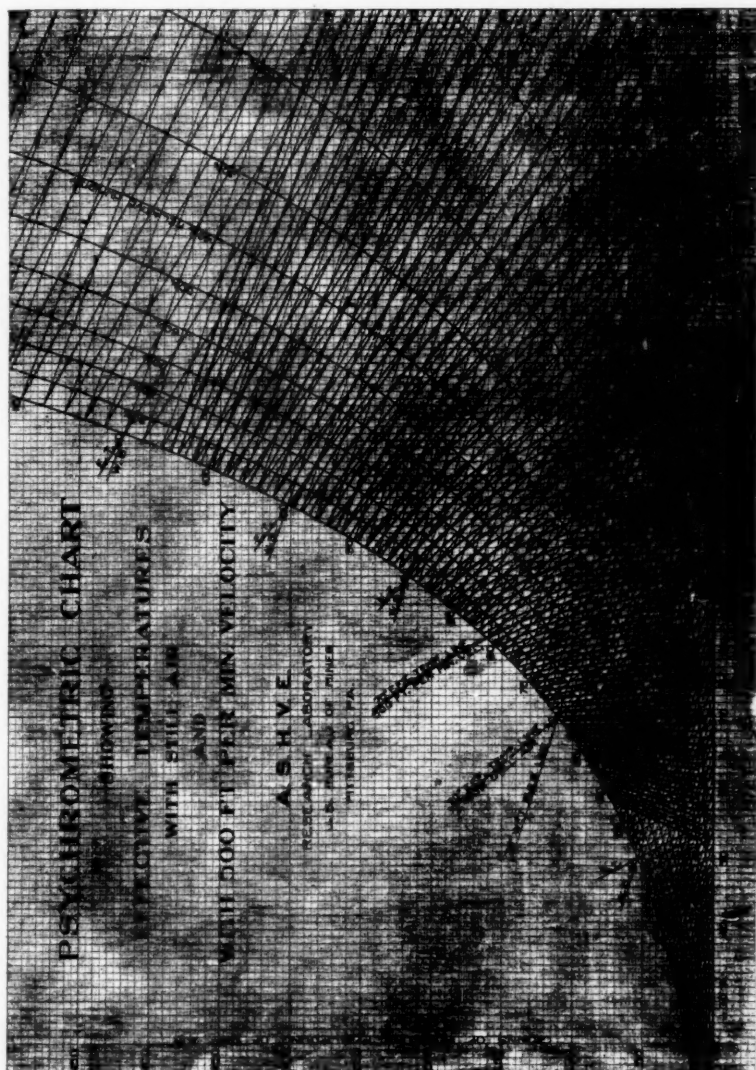


FIG. 3. PSYCHROMETRIC CHART SHOWING EFFECTIVE TEMPERATURE LINES WITH STILL AIR AND WITH 500 FT. PER MINUTE VELOCITY SUPERIMPOSED

sorbing moisture is so great that all the moisture available on the surface of the body is evaporated without the aid of air motion. At saturation the greater part of the cooling results from increased convection due to the lower temperature. As the current of air comes in contact with the body its temperature is increased to a point where it is no longer saturated, so that there is a certain amount of cooling by evaporation, even at saturation, as long as the temperature of the air remains below that of the body. These facts also explain the increasing slope of the effective temperature lines with velocity.

Variation of Cooling of Moving Air with Temperature

Considering next the variation of cooling with temperature, it can be stated, in general, that it is a function of the difference in temperature between that of the body and its atmospheric environment. As this difference in temperature decreases, cooling due to air motion also decreases, and when the two temperatures are identical the effect of the wind becomes zero. The implication here is that there are certain conditions of temperature and humidity at which air movement produces no change in the thermal state of the body. These neutral conditions are represented in Figs. 1, 2, and 3 by the upper dotted line passing through the points of intersection of the various effective temperature lines for moving air with their corresponding lines for still air.

It is of interest to note that the neutral line constitutes the border line separating the cooling zone below from the heating zone above. The magnitude of wind changes to some extent the position of the neutral line; the greater the velocity the lower its position. At saturation, when the body and air attain the same temperature, the neutral line passes a little below the 100 deg. mark, depending upon the velocity of the air. For lower humidities it becomes practically parallel to the wet-bulb lines approaching the 100 deg. wet bulb for a velocity of 150 ft. per min., and the 98 deg. for 500 ft. per min.

In general it can be stated that air motion produces cooling as long as the wet-bulb temperature is below body temperature irrespective of whether the dry bulb is 100 or 170 deg. The amount of cooling is rather small and indicates that air movement, at these high temperatures, effects an excess of heat loss by evaporation over that gained by radiation and convection. The human body, however, does not become a perfect wet bulb, and this heat gain is probably responsible for the divergence of the effective temperature lines from the wet-bulb lines.

Above the neutral line the effective temperature lines are straight, and for a velocity of 500 ft. per min. they become parallel to the wet-bulb temperature lines at an effective temperature of 115 deg., when the human body attains the thermal properties of a wet-bulb thermometer. For the lower velocities the corresponding temperatures are higher; the lower the velocity the higher the temperatures at which they become parallel to the wet-bulb lines approaching the limit of 132 deg. for still air.

Results Above Critical Temperature

Although impractical yet it is of interest to know what happens beyond this critical temperature. A few observations with 500 ft. velocity show that at an effective temperature of about 120 deg. the lines still run parallel to the wet-bulb lines. Theoretical considerations point also to this effect.

Similarly in the cool region of the chart there is a definite temperature for every air velocity at which comfort is independent of wet-bulb or relative humidity. These critical temperatures are 69, 63, 55, and 32 deg. dry bulb and correspond

to 49, 45, 40 and 32 deg. effective temperature for velocities of 500, 300, 150 and 0 ft., respectively. Below this temperature the lines slope in the opposite direction indicating a reversal in the effect of humidity. In other words, for the same dry-bulb temperature the higher the humidity the cooler the condition, and *vice versa*.

No curvature in the lines could be detected in this region of the chart, and the effect of wind is to change the slope and position of the lines. The higher the velocity the greater the slope and the higher the temperature at which dry bulb becomes the only index of comfort.

The dotted loop in the lower part of the charts constitutes another heating zone resulting from the failure of the body to supply the moisture demanded for evaporation. It was observed during the tests that within this loop the surface of the body was comparatively dry, and that the current of air effected appreciable heating on the exposed parts of the body.

From the above it is concluded that the amount of perspiration available for evaporation plays a very important part in the cooling produced by air movement and in the curvature and slope of the lines. It was observed that inducement of sweat through muscular exercise at high temperatures and very low humidities effected a marked cooling in the presence of wind, as a result of which the effective temperature lines swung upward toward the wet-bulb lines, and the lower heating zone disappeared completely.

Variation of Cooling with Velocity

The charts previously discussed show the general tendencies and characteristics of the effective temperature lines for constant velocities, and their relation to the principal physical qualities of air. To study the variation of effective temperature and cooling with velocity Figs. 4 to 8 have been prepared for constant humidities of 20, 40, 60, 80, and 100 per cent, respectively. These charts give the effective temperature with any velocity from zero to 700 ft. per min., and also the cooling both in degrees effective temperature and dry bulb, resulting from the movement of the air.

The effective temperature lines are shown by the curved lines spaced in 2-degree intervals, and are determined from the dry bulb, relative humidity, and velocity of air. The relative humidity is marked at the top of each chart. On the right is given the scale of dry-bulb temperature, while on the left, the zero velocity line constitutes the scale of effective temperature. The effective temperature of any condition is determined by passing from the point of intersection of the dry-bulb temperature and velocity parallel to the curved lines to the scale at the left. The effective temperature of the condition with still air is read on the same scale by passing horizontally from the point of intersection. The difference between the two values represents the cooling produced by the particular velocity.

To find the cooling in degrees dry bulb, follow the effective temperature line of the condition to zero velocity and read the corresponding dry-bulb temperature on the scale on the right. The difference between the dry bulb of the given condition and that found is the cooling in degrees dry-bulb temperature.

As an example of the use of the charts assume a condition of 80 deg. dry bulb, 60 per cent relative humidity, and 300 ft. velocity. From Fig. 6 it is found that the effective temperature will be 66.5 deg., the effective temperature with still air 74 deg., and the cooling produced by the movement of the air is 74 deg. - 66.5 deg. = 7.5 deg. effective temperature. The dry-bulb temperature of the equivalent

condition for still air is 71.2 deg. and the cooling in degrees dry-bulb will be 80.0 deg. - 71.2 deg. = 8.8 deg.

If it is desired to find the velocity required to reduce the original condition to 64 deg. effective temperature, pass horizontally from the dry-bulb temperature of 80 deg. to the left until the 64 deg. effective temperature line is crossed. Then read vertically down from this point of intersection the velocity value of 450 ft., to which the air movement must be increased for maximum comfort. In cases where the humidity is not represented in the charts, the values can be interpolated between the next higher and lower humidities.

The variation of cooling with velocity and temperature is shown by the change in the curvature and slope of the lines. It is of interest to note that doubling the velocity does not double the cooling, but low velocities are considerably more efficient than high. In other words, the increase of cooling with velocity is greater for low velocities and decreases as the velocity increases. Above 300 ft. per min. cooling becomes practically a straight line function of velocity. Because of this fact the effective temperature lines were extended dotted to 700 ft., although the highest velocity employed in the tests was 500 ft. per min.

Similarly, no data was obtained for velocities between zero and 150 ft., but the best fitting curve was passed through the experimental points. It will be observed, that at low temperatures and velocities there is considerable curvature in the lines as drawn. As the temperature increases the curvature becomes less pronounced until at the neutral conditions the lines become perfectly straight and horizontal indicating no change in the thermal state of the body. Above the neutral condition the slope and curvature of the lines increase in the opposite direction, indicating the heating effect of wind.

This uniform variation probably indicates that there is no abrupt change in the curvature of the lines. Furthermore, by plotting the effective temperature values for 50 ft. velocity, as obtained from the charts, against dry-bulb temperature for constant relative humidity it was found that the curves were very similar and showed a definite relation to those plotted for the velocities employed in the tests. The values given by the curves for velocities between zero and 150 ft. can therefore be taken as sufficiently accurate for practical purposes.

Table for Determining Effective Temperature

For practical use Table 1 has been prepared, from which the information given in the various charts can be obtained either directly or by interpolation. Given the dry- and wet-bulb temperature, and velocity of a condition, the relative humidity is first determined from the auxiliary table on the left. This value is then found in the proper velocity column, and the effective temperature is read by passing vertically down to the given dry-bulb temperature. In a similar manner the effective temperature for still air is found, and the difference between the two is the cooling in degrees effective temperature. The cooling in degrees dry bulb is the difference between the given dry bulb and that of the equivalent condition for still air. The latter is found by passing vertically upward from the effective temperature with still air to a value equal to that of the given condition, and reading the corresponding dry-bulb temperature in the first column of them in a table. For humidities and velocities not given in the table the data can be interpolated between the next higher and lower values.

Example: Given 85 deg. dry bulb, 74 deg. wet bulb, and 200 ft. per min. velocity,

find: 1. The effective temperature of the condition. 2. The cooling produced by air motion. 3. The velocity necessary to reduce the condition to 69 deg. effective temperature.

1. The relative humidity obtained from the auxiliary table is 60 per cent. From this value at the top of the 200 ft. velocity column pass vertically down and read 73.5 deg. effective temperature, corresponding to the dry bulb of 85 deg.

2. In a similar manner read 78 for the effective temperature with zero velocity. The desired cooling will be $78.0 \text{ deg.} - 73.5 \text{ deg.} = 4.5 \text{ deg.}$ effective temperature. From the value of 78 pass vertically upward and find 73.9 and 73.1, the nearest values to 73.5 deg. The corresponding dry-bulb temperatures are 80 deg. and 79 deg. respectively, or 79.5 deg. for an effective temperature of 73.5 deg. The cooling in degrees dry bulb will therefore be $85 \text{ deg.} - 79.5 \text{ deg.} = 5.5 \text{ deg.}$

3. From the dry-bulb temperature of 85 deg. pass horizontally to the right until 69.3 deg. effective temperature is read at the humidity column of 60 per cent. The corresponding velocity of 500 ft. found at the top of the column is the velocity necessary to reduce the original condition to 69 deg. effective temperature.

Summary of Facts

1. In general, moving air exerts a cooling effect on the human body as long as the wet-bulb temperature remains below body temperature irrespective of whether the dry bulb is 100 or 170 deg. Above body temperature air movement increases the discomfort.

2. For ordinary temperatures maximum cooling from any velocity occurs at saturation, while above the neutral line maximum heating occurs at saturation.

3. For ordinary temperatures, the higher the velocity the more predominant factor becomes the dry-bulb temperature as an index to comfort. There is a definite temperature depending upon the velocity of air at which comfort is independent of wet-bulb or relative humidity. Below this temperature the higher the humidity the cooler the condition, while above this temperature the higher the humidity the warmer the condition.

4. At high temperatures when the surface of the body is completely covered with perspiration the higher the velocity the more predominant factor wet-bulb temperature becomes as an index of comfort.

Applications

The fundamental laws of the thermal properties of the human body presented in the first part of this paper find many applications in the routine of life. During the past few years the need for artificially cooling school rooms, theaters, auditoriums, and other occupied spaces in warm weather has attracted considerable attention. Many adverse conditions such as those found in factories, mines, and steel, iron and glass works produce fatigue and affect injuriously the health of those exposed. The economic loss resulting through reduced efficiency cannot be over-emphasized, and in many instances it becomes necessary to suspend operations because of the inability of men to endure the extreme temperature conditions.

TABLE 1. AUXILIARY TABLE FOR DETERMINING RELATIVE HUMIDITY

Dry-Bulb Temp.	WET-BULB TEMPERATURE					
	Relative Humidity					
	90%	80%	70%	60%	50%	40%
30	29.0	28.0	27.0	26.0	25.3	24.2
35	34.0	33.1	31.8	30.7	29.4	28.3
40	38.9	37.6	36.3	35.0	33.8	32.4
45	43.7	42.2	40.9	39.2	38.0	36.3
50	48.6	46.9	45.3	43.6	42.0	40.2
52	50.4	48.8	47.0	45.4	43.7	41.9
54	52.4	50.7	48.9	47.1	45.3	43.3
56	54.3	52.6	50.8	48.8	47.0	45.0
58	56.3	54.5	52.4	50.5	48.6	46.5
60	58.2	56.4	54.3	52.2	50.3	48.0
61	59.2	57.3	55.2	53.1	51.1	48.8
62	60.2	58.3	56.1	54.0	52.0	49.6
63	61.1	59.2	57.1	54.9	52.8	50.3
64	62.1	60.1	58.0	55.8	53.6	51.1
65	63.1	61.0	58.9	56.7	54.4	52.0
66	64.0	62.0	59.8	57.5	55.2	52.8
67	65.0	62.9	60.6	58.3	56.0	53.6
68	66.0	63.9	61.6	59.2	56.9	54.3
69	66.9	64.9	62.4	60.1	57.7	55.0
70	67.9	65.8	63.4	61.0	58.5	55.9
71	68.8	66.8	64.2	61.8	59.3	56.7
72	69.8	67.7	65.2	62.6	60.1	57.4
73	70.8	68.5	66.1	63.5	61.0	58.2
74	71.8	69.4	66.9	64.4	61.8	59.0
75	72.8	70.4	67.9	65.3	62.5	59.8
76	73.8	71.3	68.8	66.2	63.4	60.4
77	74.8	72.2	69.7	67.0	64.2	61.2
78	75.6	73.2	70.6	67.9	65.1	62.0
79	76.7	74.1	71.4	68.8	65.9	62.8
80	77.6	75.1	72.4	69.6	66.7	63.5
81	78.6	76.0	73.3	70.4	67.5	64.3
82	79.6	77.0	74.3	71.3	68.3	65.1
83	80.4	78.0	75.2	72.2	69.2	65.9
84	81.4	78.8	76.1	73.1	70.0	66.6
85	82.3	79.8	77.1	74.0	70.8	67.3
86	83.3	80.7	78.0	74.9	71.6	68.1
87	84.3	81.6	78.9	75.7	72.4	68.9
88	85.2	82.6	79.8	76.6	73.3	69.8
89	86.2	83.6	80.7	77.4	74.2	70.4
90	87.2	84.5	81.6	78.4	75.0	71.2
92	89.2	86.4	83.3	80.1	76.6	72.9
94	91.2	88.3	85.2	81.8	78.3	74.4
96	93.2	90.2	87.0	83.7	80.0	76.0
98	95.2	92.2	88.8	85.4	81.6	77.7
100	97.2	94.0	90.6	87.1	83.2	79.2
102	99.0	95.9	92.5	88.9	84.9	80.8
104	101.0	97.8	94.3	90.6	86.6	82.3
106	102.9	99.6	96.2	92.4	88.3	83.9
108	104.9	101.4	98.0	94.2	90.0	85.4
110	106.9	103.3	99.8	95.9	91.7	87.0
112	108.9	105.2	101.6	97.7	93.3	88.6
114	110.8	107.1	103.4	99.3	95.0	90.2
116	112.7	109.0	105.3	101.1	96.7	91.8
118	114.6	111.0	107.2	102.9	98.4	93.4
120		113.0	109.1	104.7	100.1	95.0
125			113.7	109.2	104.3	98.9
130				113.7	108.6	102.8
135					112.8	106.8
140					117.0	110.8
145						114.8
150						111.2
						101.7

TABLE 1A. DETERMINING EFFECTIVE TEMPERATURE

Dry-Bulb Temp.	EFFECTIVE TEMPERATURE Velocity of Air in Feet per Minute									
	STILL AIR					50 Fr.				
	Relative Humidity					Relative Humidity				
	100%	80%	60%	40%	20%	100%	80%	60%	40%	20%
30	30	30.2	30.3	30.4	30.6					
35	35	34.9	34.7	34.5	34.4					
40	40	39.6	39.2	38.8	38.4	31.0	31.2	31.3	31.5	31.8
45	45	44.3	43.7	43.1	42.5	37.2	36.8	36.5	36.2	36.0
50	50	49.1	48.2	47.3	46.6	43.4	42.6	42.0	41.6	41.4
52	52	51.0	50.0	49.0	48.1	45.9	45.0	44.3	43.8	43.5
54	54	52.9	51.8	50.7	49.7	48.3	47.2	46.4	45.7	45.3
56	56	54.8	53.4	52.3	51.2	50.8	49.8	48.8	47.9	47.3
58	58	56.6	55.2	54.0	52.8	53.1	51.9	50.9	50.0	49.1
60	60	58.5	57.0	55.5	54.2	55.4	54.1	53.0	52.0	50.9
61	61	59.4	57.8	56.4	55.0	56.7	55.2	54.0	52.8	51.8
62	62	60.3	58.7	57.2	55.7	57.9	56.3	55.0	53.8	52.6
63	63	61.2	59.5	58.0	56.4	59.0	57.3	55.9	54.6	53.5
64	64	62.1	60.4	58.8	57.2	60.1	58.3	56.9	55.5	54.3
65	65	63.0	61.3	59.6	57.9	61.2	59.4	57.8	56.3	55.1
66	66	64.0	62.2	60.3	58.6	62.4	60.6	58.8	57.2	56.0
67	67	65.0	63.0	61.1	59.3	63.6	61.7	59.9	58.1	56.9
68	68	65.9	63.8	61.9	60.0	64.7	62.8	61.0	59.0	57.8
69	69	66.9	64.7	62.7	60.7	66.0	63.9	62.0	60.0	58.7
70	70	67.8	65.5	63.4	61.3	67.1	64.9	62.9	61.0	59.3
71	71	68.7	66.3	64.2	62.0	68.2	65.9	63.8	61.9	60.1
72	72	69.6	67.2	65.0	62.8	69.4	66.9	64.7	62.8	60.9
73	73	70.5	68.0	65.7	63.4	70.6	68.0	65.8	63.7	61.8
74	74	71.4	68.9	66.5	64.1	71.7	68.9	66.7	64.4	62.4
75	75	72.3	69.8	67.2	64.8	72.8	69.9	67.6	65.2	63.1
76	76	73.2	70.5	68.0	65.3	73.9	71.0	68.5	66.0	63.9
77	77	74.2	71.4	68.7	66.1	75.0	72.1	69.5	66.9	64.6
78	78	75.1	72.2	69.5	66.8	76.2	73.1	70.3	67.9	65.3
79	79	76.0	73.1	70.2	67.4	77.4	74.0	71.3	68.7	66.0
80	80	77.0	73.9	71.0	68.0	78.6	75.0	72.1	69.4	66.8
81	81	77.8	74.7	71.7	68.7	79.8	75.9	73.0	70.1	67.5
82	82	78.7	75.5	72.4	69.4	80.9	77.0	74.0	70.8	68.2
83	83	79.6	76.3	73.2	70.0	82.1	78.0	74.8	71.5	68.9
84	84	80.5	77.2	74.0	70.7	83.1	79.0	75.8	72.4	69.6
85	85	81.4	78.0	74.7	71.3	84.2	80.0	76.6	73.3	70.2
86	86	82.3	78.8	75.3	72.0	85.3	81.0	77.5	74.0	70.9
87	87	83.2	79.6	76.0	72.6	86.3	82.0	78.3	74.8	71.8
88	88	84.2	80.4	76.8	73.2	87.3	83.0	79.1	75.6	72.3
89	89	85.1	81.3	77.4	73.8	88.4	84.0	80.0	76.4	73.0
90	90	86.0	82.0	78.2	74.3	89.4	84.8	80.8	77.1	73.7
92	92	87.8	83.7	79.6	75.5	91.5	86.7	82.3	78.4	74.8
94	94	89.6	85.3	81.0	76.8	93.7	88.7	84.1	80.0	76.0
96	96	91.5	87.0	82.4	78.0	95.9	90.6	85.9	81.3	77.3
98	98	93.3	88.6	83.9	79.2	98.0	92.6	87.6	82.7	78.6
100	100	95.2	90.2	85.2	80.2	100.0	94.6	89.2	84.2	79.8
102	102	96.9	91.8	86.7	81.3	102.0	96.6	91.0	85.8	80.9
104	104	98.7	93.4	88.0	82.4	104.2	98.6	92.8	87.2	82.0
106	106	100.4	95.0	89.4	83.5	106.3	100.4	94.4	88.6	83.2
108	108	102.2	96.6	90.9	84.6	108.4	102.3	96.0	90.0	84.3
110	110	104.1	98.1	92.2	85.7	110.7	104.3	97.9	91.6	85.3
112	112	106.0	99.8	93.7	86.9	113.0	106.3	99.8	92.9	86.5
114	114	107.8	101.3	95.0	88.0	115.3	108.3	101.4	94.2	87.7
116	116	109.6	102.9	96.3	89.1		110.3	103.2	95.8	88.8
118	118	111.4	104.6	97.5	90.2		112.3	104.8	97.2	90.0
120	120	113.3	106.2	98.9	91.3		114.5	106.7	98.6	91.0
125			110.3	102.3	94.0			111.0	102.0	93.8
130			114.5	105.7	96.4			115.7	105.9	96.2
135				109.2	99.0				109.7	98.8
140				112.6	101.6				113.6	101.6
145				116.1	104.2					104.1
150					106.7					106.9

TABLE 1A. DETERMINING EFFECTIVE TEMPERATURE—(Continued)

Dry-Bulb Temp.	EFFECTIVE TEMPERATURE									
	Velocity of Air in Feet per Minute									
	100 Fr.					200 Fr.				
	Relative Humidity					Relative Humidity				
	100%	80%	60%	40%	20%	100%	80%	60%	40%	20%
30										
35										
40										
45	30.7	31.0	31.3	31.6	31.9					
50	37.3	37.2	37.2	37.1	37.1					
						30.0	30.8	31.5	32.0	
52	40.1	40.0	39.8	39.6	39.4	32.0	32.8	33.5	34.0	34.4
54	43.0	42.5	42.1	41.9	41.7	35.2	35.6	36.1	36.4	36.8
56	45.7	45.0	44.4	44.1	43.8	38.3	38.6	38.8	39.0	39.2
58	48.3	47.6	46.9	46.3	45.9	41.3	41.4	41.5	41.6	41.6
60	51.0	50.0	49.3	48.6	48.0	44.4	44.3	44.2	44.1	44.0
61	52.5	51.3	50.5	49.8	49.0	46.0	45.6	45.3	45.2	45.1
62	53.8	52.7	51.7	50.7	49.9	47.5	47.0	46.7	46.4	46.2
63	55.1	53.9	52.9	51.9	50.9	49.1	48.5	48.0	47.7	47.3
64	56.4	55.1	54.0	52.9	51.9	50.7	50.0	49.3	48.8	48.4
65	57.7	56.2	55.0	53.9	52.8	52.2	51.3	50.7	50.0	49.5
66	59.0	57.4	56.1	54.9	53.8	53.7	52.8	51.9	51.1	50.6
67	60.4	58.6	57.2	55.9	54.7	55.2	54.0	53.1	52.3	51.7
68	61.7	60.0	58.3	56.8	55.7	56.8	55.5	54.4	53.6	52.8
69	63.0	61.2	59.4	57.8	56.7	58.3	56.8	55.8	54.8	53.8
70	64.3	62.3	60.5	58.8	57.6	59.9	58.0	56.8	55.9	54.8
71	65.6	63.6	61.8	60.0	58.5	61.4	59.4	58.0	57.0	55.9
72	66.8	64.6	62.6	60.8	59.4	62.9	60.7	59.1	58.0	56.9
73	68.1	65.8	63.7	61.8	60.1	64.4	62.1	60.2	59.0	57.0
74	69.4	66.8	64.7	62.6	61.0	65.8	63.6	61.6	60.0	58.9
75	70.7	67.9	65.8	63.6	61.8	67.3	64.9	62.7	61.1	59.8
76	72.0	69.0	66.8	64.4	62.6	68.8	66.0	64.0	62.2	60.7
77	73.2	70.1	67.9	65.4	63.4	70.3	67.2	65.0	63.2	61.8
78	74.5	71.5	68.8	66.3	64.0	71.7	68.5	66.1	64.1	62.4
79	75.8	72.4	69.8	67.2	64.9	73.1	69.8	67.3	65.2	63.4
80	77.1	73.6	70.7	68.0	65.8	74.5	70.9	68.3	66.0	64.2
81	78.4	74.5	71.8	68.9	66.4	75.9	72.2	69.4	67.0	65.2
82	79.6	75.8	72.8	69.8	67.2	77.3	73.4	70.3	68.0	66.0
83	80.9	76.9	73.6	70.6	68.0	79.7	74.7	71.5	68.9	66.9
84	82.1	78.0	74.4	71.4	68.8	80.1	75.9	72.5	69.8	67.8
85	83.4	79.0	75.4	72.2	69.6	81.5	76.9	73.5	70.8	68.4
86	84.6	80.0	76.3	72.8	70.3	82.9	78.0	74.5	71.7	69.3
87	85.7	81.0	77.3	73.8	71.0	84.2	79.3	75.7	72.4	70.0
88	86.8	82.0	78.2	74.6	71.7	85.5	80.3	76.6	73.2	70.9
89	87.9	83.0	79.0	75.4	72.3	86.8	81.6	77.6	74.2	71.7
90	89.0	84.0	79.9	76.1	73.0	88.0	82.7	78.4	75.1	72.2
92	91.2	86.9	81.5	77.8	74.3	90.4	84.9	80.2	76.8	73.8
94	93.4	88.0	83.3	79.2	75.6	92.8	87.1	82.3	78.3	75.0
96	95.6	90.0	85.0	80.8	76.8	95.3	89.3	84.0	80.0	76.4
98	97.8	92.2	86.8	82.1	78.0	97.7	91.6	86.0	81.6	77.8
100	100.0	94.2	88.6	83.8	79.3	100.0	93.8	87.8	83.1	79.0
102	102.2	96.4	90.2	85.2	80.6	102.4	96.0	89.7	84.7	80.3
104	104.4	98.4	92.1	86.7	81.8	105.1	98.2	91.5	86.1	81.7
106	106.7	100.3	94.0	88.0	82.9	107.7	100.5	93.4	87.7	82.8
108	109.1	102.4	95.8	89.6	84.0	110.5	102.6	95.4	89.1	83.9
110	111.6	104.5	97.7	90.9	85.2	113.9	105.1	97.4	90.5	85.0
112	114.1	106.6	99.6	92.5	86.3		107.3	99.6	92.0	86.2
114		108.8	101.4	94.0	87.5		110.0	101.4	93.6	87.3
116		111.0	103.2	95.3	88.6		112.6	103.4	95.2	88.4
118		113.2	105.0	96.9	89.8		115.5	105.7	96.7	89.6
120		115.8	106.9	98.4	90.9			107.8	98.1	90.7
125			111.8	102.0	93.6		113.5	102.2	93.3	
130			106.0	96.0	87.7			106.3	95.8	
135			110.0	98.7				111.2	98.2	
140			114.4	101.4				116.2	101.2	
145				104.2					104.2	
150				107.1					107.4	

TABLE 1A. DETERMINING EFFECTIVE TEMPERATURE—(Continued)

EFFECTIVE TEMPERATURE Velocity of Air in Feet per Minute											
300 Fr.						500 Fr.					
100%	80%	Relative Humidity	60%	40%	20%	100%	80%	Relative Humidity	60%	40%	20%
30.1											
30.3	31.8	30.4	31.5	32.3							
33.6	34.8	33.0	34.0	34.7							
36.9	37.8	35.8	36.5	37.0							
40.2	40.8	41.2	41.4	41.6	30.0	32.0	31.0	30.0	31.2	33.8	33.8
41.9	42.2	42.4	42.8	42.9	35.2	36.9	38.1	39.0	40.0		
43.4	43.6	43.8	44.0	44.1	36.9	38.2	39.4	40.4	41.1		
45.0	45.0	45.0	45.0	45.0	38.6	39.9	41.0	41.7	42.2		
46.7	46.4	46.2	46.1	46.0	40.4	41.4	42.3	43.0	43.3		
48.3	48.0	47.7	47.4	47.1	42.2	43.0	43.8	44.2	44.5		
49.9	49.4	49.0	48.7	48.3	44.0	44.6	45.0	45.4	45.7		
51.5	50.8	50.3	49.9	49.4	45.8	46.1	46.4	46.7	46.9		
53.1	52.2	51.6	51.0	50.6	47.5	47.7	47.8	48.0	48.1		
54.7	53.8	53.0	52.2	51.7	49.1	49.1	49.1	49.1	49.1		
56.3	55.2	54.2	53.4	52.8	51.0	50.8	50.5	50.2	50.0		
58.0	56.6	55.6	54.7	53.9	52.8	52.2	51.9	51.4	51.1		
59.6	58.0	56.8	55.9	55.0	54.5	53.7	53.0	52.6	52.3		
61.2	59.3	58.0	56.9	56.0	56.2	55.1	54.3	53.9	53.3		
62.8	60.8	59.1	58.0	57.0	58.0	56.8	55.8	55.0	54.3		
64.3	62.1	60.4	59.0	58.0	59.7	58.1	57.0	56.1	55.3		
65.8	63.6	61.7	60.1	59.0	61.4	59.6	58.3	57.2	56.4		
67.4	64.9	63.0	61.3	60.0	63.2	61.0	59.7	58.4	57.5		
69.0	66.1	64.1	62.3	61.0	64.9	62.5	60.8	59.7	58.6		
70.6	67.4	65.2	63.4	61.9	66.5	63.9	62.0	60.7	59.7		
72.1	68.9	66.4	64.6	62.9	68.2	65.3	63.2	61.8	60.7		
73.6	70.1	67.6	65.6	63.9	69.9	66.7	64.4	62.9	61.6		
75.1	71.4	68.9	66.6	64.8	71.5	68.1	65.7	64.0	62.5		
76.6	72.7	69.9	67.7	65.8	73.2	69.4	66.9	65.1	63.4		
78.1	74.0	71.0	68.7	66.7	74.9	70.9	68.0	66.1	64.4		
79.6	75.2	72.0	69.6	67.5	76.5	72.2	69.3	67.1	65.4		
81.0	76.5	73.1	70.6	68.3	78.2	73.7	70.4	68.1	66.3		
82.4	77.8	74.1	71.4	69.2	79.9	75.0	71.7	69.2	67.3		
83.9	79.0	75.3	72.3	70.0	81.3	76.3	72.9	70.2	68.2		
85.3	80.2	76.3	73.3	70.8	83.0	77.7	74.0	71.3	69.1		
86.7	81.4	77.3	74.1	71.7	84.5	79.1	75.2	72.3	70.0		
89.5	83.8	79.3	76.0	73.2	87.8	81.6	77.3	74.3	71.9		
92.2	86.1	81.3	77.8	74.7	90.9	84.3	79.6	76.3	73.6		
95.0	88.4	83.3	79.3	76.2	94.0	87.0	81.7	78.0	75.3		
97.6	90.9	85.3	81.0	77.6	97.2	89.8	83.8	79.8	76.9		
100.2	93.3	87.2	82.6	79.0	100.3	92.2	85.9	81.4	78.3		
102.8	95.8	89.1	84.2	80.2	103.8	95.0	88.1	83.2	79.7		
105.6	98.2	91.0	85.8	81.4	107.4	97.7	90.1	84.9	81.1		
108.7	100.6	93.0	87.2	82.7	111.7	100.8	92.2	86.6	82.4		
112.1	103.0	95.1	88.7	83.9	116.3	103.7	94.3	88.2	83.7		
115.8	105.4	97.1	90.2	85.1		107.0	96.7	89.7	84.8		
	108.1	99.1	91.7	86.2		110.6	99.2	91.3	86.0		
	111.1	101.3	93.3	87.2		114.3	101.7	92.8	87.2		
	114.4	103.6	94.9	88.3			104.3	94.3	88.3		
		105.8	96.3	89.4			107.1	96.0	89.3		
		108.2	98.0	90.4			110.1	97.9	90.3		
		115.0	102.3	93.0				102.7	92.8		
			107.0	95.5				108.0	95.3		
			112.2	98.1				114.0	98.0		
				101.0					101.0		
				104.2					104.4		
				107.7					108.1		

In cases where air motion produces considerable cooling it is the simplest and most inexpensive method available. At high temperatures, however, the benefit derived from movement of the air is small, and steps should be taken to reduce the effective temperature by other means prior to setting the air in motion.

One of the most important principles of air conditioning is manifested in the cooling produced by the evaporation of water. When air partly saturated comes in contact with water, as for instance by passing it through a humidifier, a certain amount of heat is absorbed from the air in the process of evaporation, effecting an appreciable lowering in the temperature of the air. The wet-bulb temperature

TABLE 1A. DETERMINING EFFECTIVE TEMPERATURE (Concluded)
EFFECTIVE TEMPERATURE
Velocity of Air in Feet per Minute
700 Ft.

Dry-Bulb Temp.	Relative Humidity				Dry-Bulb Temp.	Relative Humidity			
	100%	80%	60%	40%		100%	80%	60%	40%
30					81	65.8	63.4	61.8	60.7
35					82	67.6	64.8	62.7	61.7
40					83	69.4	66.3	63.9	62.8
45					84	71.1	67.8	65.4	64.0
50					85	72.9	69.2	66.7	65.0
52					86	74.6	70.7	68.0	66.1
54					87	76.4	72.1	69.0	67.3
56					88	78.1	73.4	70.0	68.3
58					89	80.0	74.9	71.1	69.6
60			32.4	34.3	90	81.9	76.2	72.3	70.4
61		31.0	33.9	36.0	92	85.7	79.2	74.7	72.3
62	30.8	32.8	35.4	37.1	94	89.5	82.0	77.2	74.2
63	32.7	34.4	36.7	38.4	96	93.3	84.6	79.7	76.2
64	34.6	36.0	38.2	40.0	98	97.0	87.6	82.0	78.2
65	36.5	38.0	39.8	41.2	100	100.8	90.8	81.2	80.2
66	38.3	39.7	41.2	42.6	102	104.8	94.0	86.5	82.2
67	40.2	41.3	42.5	44.0	104	108.8	97.2	88.8	84.0
68	42.0	42.9	44.0	45.3	106	112.8	100.0	91.1	85.8
69	44.0	44.7	45.6	46.6	108		104.3	93.8	87.7
70	46.0	46.4	47.0	47.8	110		108.3	96.2	89.3
71	48.0	47.8	48.3	49.1	112		112.6	98.9	90.8
72	49.9	49.9	50.0	50.0	114			101.8	92.6
73	51.8	51.6	51.4	51.2	116			104.8	94.2
74	53.6	52.9	52.7	52.5	118			107.8	96.0
75	55.4	54.6	54.2	53.8	120			111.1	97.8
76	57.2	55.9	55.3	54.8	125				103.0
77	58.9	57.3	56.7	56.1	130				109.4
78	60.6	58.9	58.0	57.2	135				115.5
79	62.3	60.3	59.3	58.5	140				
80	64.0	62.0	60.5	59.6	145				
					150				

remains the same if no heat is added or subtracted from the system, the sensible heat of the air being transformed to latent heat. Ultimately the dry-bulb temperature of the air is reduced to that of the wet bulb when the air becomes completely saturated.

Referring to Fig. 2 assume a condition of 110 deg. dry bulb and 85 deg. wet bulb. The effective temperature is 91 deg., and it will be found that a velocity of 300 ft. per min. produces a cooling of only 2 deg. By passing this air through a humidifier where the water is re-circulated and not heated, we pass to the left of the chart along the 85 deg. wet-bulb line until saturation is reached. The dry-bulb temperature of the air is reduced from 110 deg. to 85 deg., and its effective temperature from 91 deg. to 85 deg. effecting a cooling of 25 deg. dry bulb and 6 deg. effective temperature. A velocity of 300 ft. now applied will produce a further cooling of 5.5 deg. effective temperature on the human body at rest and stripped to the waist as compared with 2 deg. prior to saturation, and the original condition of 91 deg.

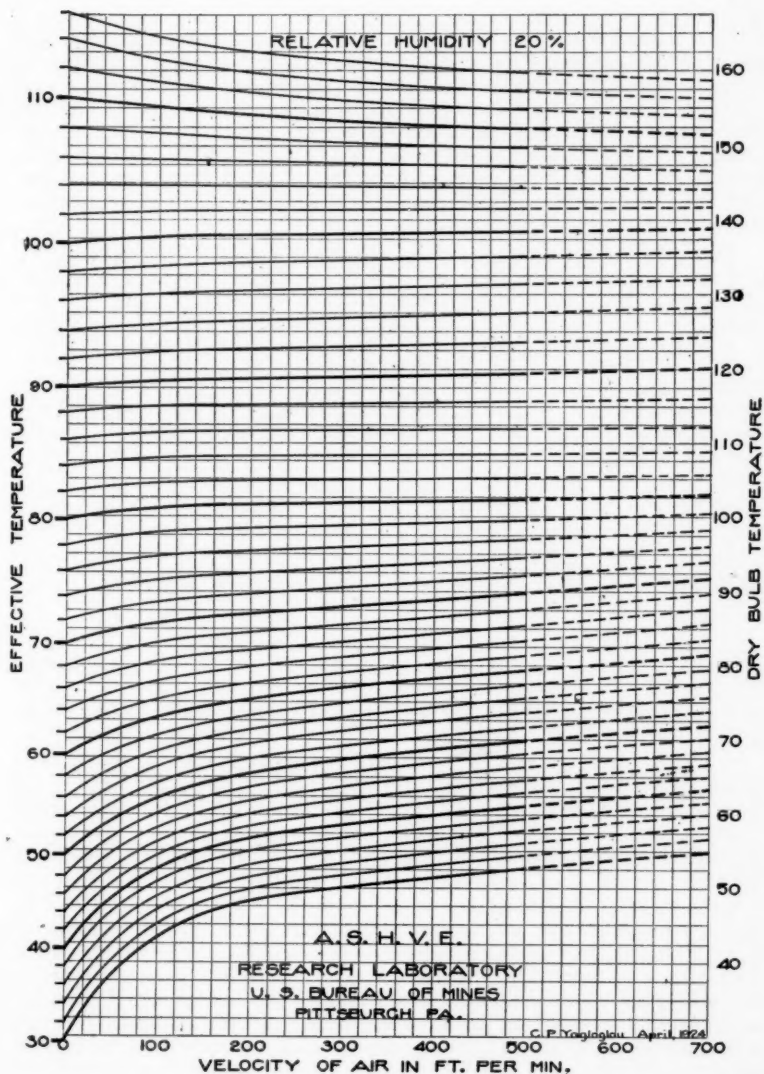


FIG. 4. VARIATION OF EFFECTIVE TEMPERATURE AND COOLING PRODUCED BY AIR MOVEMENT ACCORDING TO THE VELOCITY OF AIR, FOR CONDITIONS WITH 20% RELATIVE HUMIDITY

effective temperature is thus theoretically reduced to 79.5 deg. effective temperature.

It is of interest to note that in addition to the cooling obtained from the evaporation of water, the effect of air movement reaches a maximum value at saturation, and an enormous amount of cooling results from the combination of the two. This method of artificial cooling is very promising to many hot operations in industries where the humidity of the air is not very high. The air is simply saturated and blown upon the workers. The process involves the use of simple and inexpensive equipment as compared with refrigeration, such as humidifiers and fans, and it is very likely that this most efficient method of artificial cooling will be extensively used in the future.

It is a well-known fact that there is an optimum temperature of the environment most conducive to human comfort, and at which the body works most efficiently.

TABLE 2. GIVING COMFORTABLE CONDITIONS, IN TERMS OF DRY- AND WET-BULB TEMPERATURE, FOR NORMALLY CLOTHED PEOPLE AT REST, IN STILL AND MOVING AIR

Relative Humidity	Still Air 0 ft.		50 ft.		Velocity of Air in Feet per Minute								500 ft.	
	D. B.	W. B.	D. B.	W. B.	100 ft.	100 ft.	200 ft.	200 ft.	300 ft.	300 ft.	300 ft.	300 ft.	D. B.	W. B.
100%	64.0	64.0	67.3	67.3	69.9	69.9	72.8	72.8	74.8	74.7	77.5	77.5		
90%	65.0	63.0	68.2	66.1	70.7	68.5	73.6	71.3	75.5	73.2	78.3	76.0		
80%	66.0	62.0	69.1	65.0	71.4	67.0	74.3	69.8	76.3	71.5	79.1	74.2		
70%	67.1	60.7	70.1	63.4	72.3	65.4	75.1	68.0	77.1	69.7	79.9	72.2		
60%	68.2	59.4	71.1	61.9	73.3	63.8	76.0	66.1	77.9	67.7	80.6	70.2		
50%	69.4	58.0	72.3	60.3	74.3	62.0	77.0	64.2	78.7	65.6	81.4	67.8		
40%	70.7	56.4	73.4	58.5	75.4	60.0	77.9	62.0	79.5	63.1	82.1	65.1		
30%	72.2	54.5	74.7	56.4	76.5	57.6	78.8	59.2	80.3	60.3	82.8	62.0		
20%	73.8	52.4	76.0	53.9	77.8	55.0	79.7	56.2	81.1	57.1	83.6	58.6		

This optimum temperature depends largely upon the nature of the work performed. The heat produced through the chemical changes within the body must be lost to the outside for the body temperature to remain constant. The greater the muscular activity the greater the heat produced, and the cooling power of the air should be correspondingly increased to effect a greater heat loss from the human body.

The optimum temperature for individuals at rest, or otherwise engaged in light activities, in still air and normally clothed was found by the Research Laboratory⁴ to be 64.5 deg. effective temperature. In round figures an effective temperature of 64 deg. should be adopted for dwellings, office buildings, theaters, schools and all other places where mental and light muscular work is performed in practically still air.

Table 2 gives the conditions corresponding to the effective temperature of 64 deg. in terms of dry bulb and wet bulb with still air and their variation with velocity. Although only the data for still air was determined experimentally by a great variety of individuals, a comparison of a number of equivalent conditions with still air and with 500 ft. velocity, made by a few persons normally clothed, disclosed no definite variation. While all equivalent conditions are equally comfortable, practical considerations limit the range of humidity to be ordinarily employed from 30 to 70 per cent. As a general rule relative humidities from 50 to 60 per cent will be found most desirable.

Subsequent experiments will determine the optimum temperature for individuals

⁴ Determination of the Comfort Zone and Further Verification of Effective Temperature within this Zone, by F. C. Houghton and C. P. Yaglou, JOURNAL A.S.H.&V.E., September, 1923.

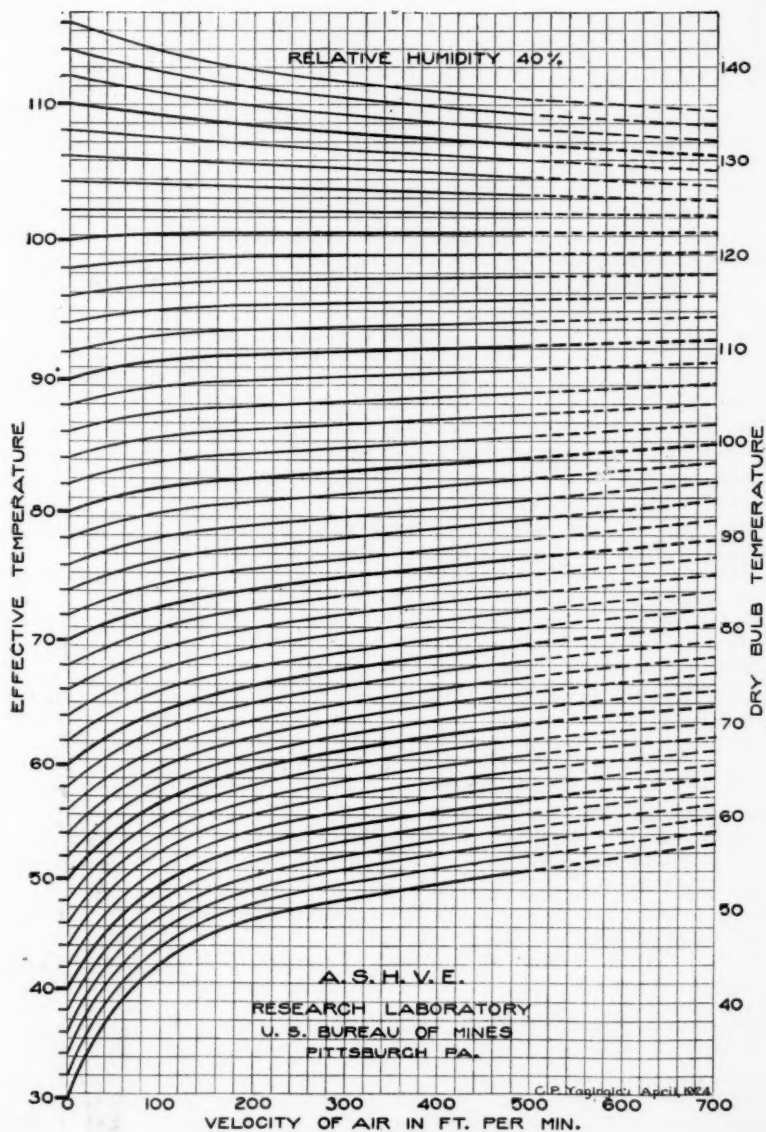


FIG. 5. VARIATION OF EFFECTIVE TEMPERATURE AND COOLING PRODUCED BY AIR MOVEMENT ACCORDING TO THE VELOCITY OF AIR, FOR CONDITIONS WITH 40% RELATIVE HUMIDITY

doing various amounts of work both in still and in moving air of different velocities, so that a standard can be set for every representative industry.

Application to Factories and Workshops

The influence of atmospheric conditions in factories and workshops upon the health and well-being of the occupants cannot be overlooked. The industrial worker spends the major part of his active life in an environment where heat, moisture, and in some special cases, injurious elements are constantly evolved by the process of manufacture. These conditions, as a result of their effect on the health and comfort of the workers, are chiefly responsible for the quantity and quality of output and, therefore, for the general efficiency of the plant.

The Industrial Fatigue Board of England, in a report of its extensive investigation on the fatigue and efficiency of industrial workers, states that for temperatures under 40 deg. fahr. the hourly output was 10 per cent above normal. Conversely, a rise in temperature was followed by a decrease in efficiency, and for an external temperature of 65 deg. the hourly output was 10 per cent below normal. The maximum seasonal fluctuation in output observed was about 30 per cent below normal and occurred in the summer months.

For the native American these temperatures are rather low. Various observations show that the European standard of temperature for comfort is considerably lower than that of America.

In a report of the importance of temperature and humidity to physical work the New York State Commission on Ventilation states that men perform 28 per cent less physical work in a temperature of 86 deg. fahr. with 80 per cent relative humidity than in one of 68 deg. fahr. and 50 per cent relative humidity. An estimate of the daily loss in output in an average size industrial plant operating at an efficiency of 70 per cent will reveal the material financial loss resulting from failure to control the temperature conditions.

It is a comparatively simple matter to produce and maintain proper atmospheric conditions indoors in winter. In summer, however, with an outside temperature in the neighborhood of 95 deg., the problem becomes rather complicated. The incoming air diffusing into the workrooms takes up heat liberated from the machinery in operation and from the bodies of the workers, and its temperature is increased considerably.

While every factory is equipped with a heating system, little provision is made for cooling during the hot summer months, despite the fact that the greatest seasonal fluctuations in efficiency occur in summer.

It has been shown in general how high temperature conditions can be improved by means of saturation and air movement. As an actual application of the method, the case of an automobile factory is taken, where the average summer observations available for 1923 were about 96 deg. dry bulb and 80 deg. wet bulb, with practically still air. From the charts or table it will be found that this condition corresponds to an effective temperature of 84.7 deg., and that a velocity of 300 ft. will improve the situation by only 3.5 deg. effective temperature. Saturating the air, however, the dry-bulb temperature is reduced to 80 deg., and a velocity of 300 ft. now applied will theoretically reduce the condition to about 72 deg. effective temperature producing a total improvement of 84.7 deg. - 72.0 deg. = 12.7 deg. effective temperature.

In practice these theoretical values cannot be attained. Allowance must be made for the increase in temperature and decrease in the humidity of the air in

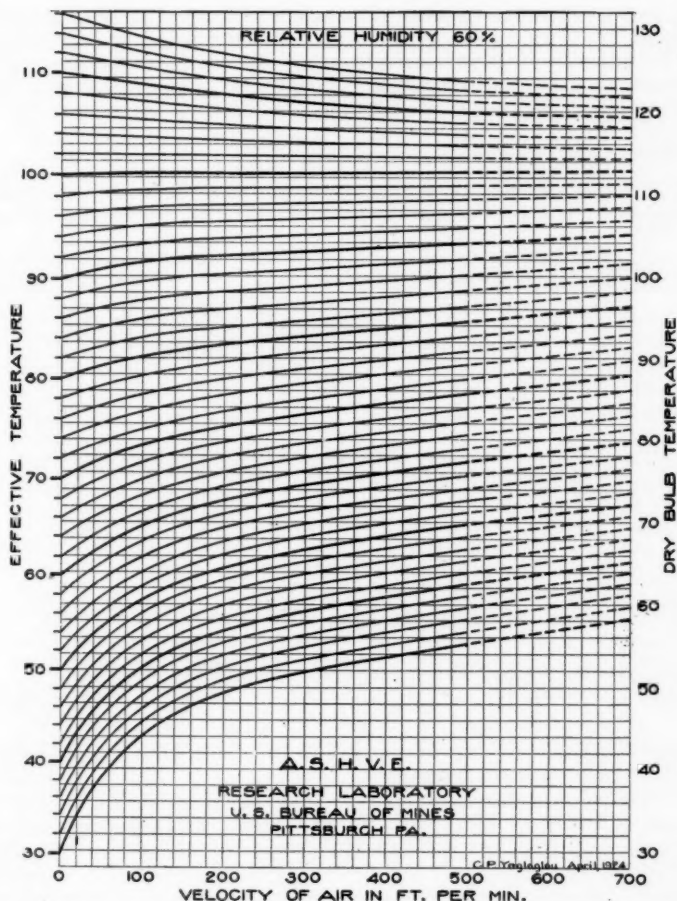


FIG. 6. VARIATION OF EFFECTIVE TEMPERATURE AND COOLING PRODUCED BY AIR MOVEMENT ACCORDING TO THE VELOCITY OF AIR, FOR CONDITIONS WITH 60% RELATIVE HUMIDITY

diffusing into the workroom before striking the bodies of the workers. In addition the values will probably be affected by the clothing worn, and the type of work performed.

In large factories the process requires the use of humidifiers and blowers, the latter forcing through ducts the cool air directly upon the workers. In addition to the cooling effect, a fresh supply of air is provided at all times to remove the products of respiration and various other injurious elements evolved by the process of manufacture.

In small factories and workshops the desired cooling power of the air can be obtained by locally applied electric fans directing a current of air upon the workers. The velocity of the air being comparatively high a few feet from the fans the cooling produced by the wind alone will be sufficient in the majority of cases, without the use of humidifiers

Application to Mining Industry

Until recently the problem of mine ventilation from the standpoint of maintaining proper purity of the air underground was given considerable attention

Dry Kata Cooling power	1	2	3	4	5	6
Working efficiency per cent	50	60	70	80	90	100

from the part of many investigators. However, little was accomplished in the way of providing better working conditions essential to the well-being and efficiency of miners.

Adverse conditions of temperature and humidity in many mines impair the health and vitality of the workers and lower the output, with the consequent immense financial loss to the owners of the mines. Orenstein and Ireland⁵ estimated by means of a dynamometer and ergometer the working efficiency of miners on the Rand. Their results, given below, are based on dry Kata cooling powers with the

TABLE 3

Mine	D. B. temp.	W. B. temp.	Vel. ft. per min.	E. T.	Remarks of workers
Royal Lode	88.7	88.0	0	88.5	Unendurable
Royal Lode	88.0	84.7	360	79.8	Good for work
Royal Lode	87.0	83.0	710	72.5	Excessive velocity
Waihi Mines, 1450 ft. level	81.4	80.2	0	80.7	Oppressive
Waihi Mines, 1450 ft. level	80.2	78.6	770	64.0	Excessive velocity

assumption that the working efficiency is 100 per cent at a dry Kata power of 6 millicalories per square cm. per second.

It will be observed that the working efficiency decreased when the Kata power fell below 6 until at 1 millicalorie per sq. cm. per sec., the average efficiency was only 50 per cent, while the bodily temperature rose considerably and extreme fatigue was produced by work. Many similar cases can be given where the temperature conditions underground were such that the miners could remain at work only half of the time, spending the other half in cooling off and rest.

In a study of the physiological effects of high temperatures and high humidities in metal mines, Sayers and Harrington⁶ found that in a still atmosphere of 80 to 90 deg. Fahr., manual labor caused the body temperature to rise in a short period to 102 deg., and frequently to 103 deg. when the relative humidity exceeded 95 per cent. The pulse rate increased rapidly, accompanied by physical weakness, exhaustion, inability to think, and a marked loss in weight. However, with air velocities of between 400 and 500 ft. per minute, there was no marked change in body temperature or pulse rate, and no appreciable discomfort was experienced.

⁵ A Contribution to the Study of the Influence of Mine Atmospheric Conditions on Fatigue, by A. J. Orenstein and H. J. Ireland, *Journal of the South African Institution of Engineers*, March, 1921.

⁶ Physiological Effects of High Temperature and Humidity with and without Air Motion, by R. R. Sayers and D. Harrington, U. S. Bureau of Mines, Report of Investigation, Serial No. 2464.

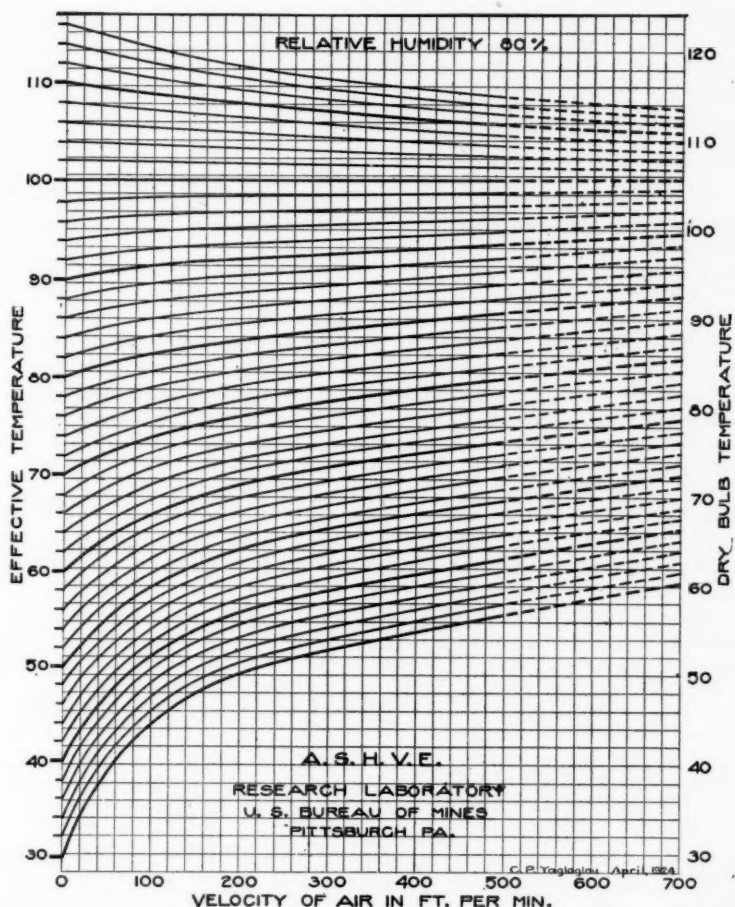


FIG. 7. VARIATION OF EFFECTIVE TEMPERATURE AND COOLING PRODUCED BY AIR MOVEMENT ACCORDING TO THE VELOCITY OF AIR FOR CONDITIONS WITH 80% RELATIVE HUMIDITY

Referring to Fig. 3, it will be found that a velocity of 500 ft. per min. at a temperature of 90 deg. dry bulb and 95 per cent relative humidity produces a cooling of 6.5 effective temperature on the human body. The original condition corresponds to 88 deg. effective temperature, while that effected by the movement of the air is 81.5 deg. effective temperature, a rather comfortable temperature for men accustomed to work in mines.

It is obvious from the above that the cooling effect of moving air is of utmost importance in the mining industry. In places where the temperature and humid-

ity cannot be readily reduced, or in small or shallow mines where the investment involved in the installation of a mechanical system of ventilation is not justified, the velocity of the air may be increased by means of locally applied fans to produce the desired cooling on the workers.

Table 3, compiled by Lewis⁷ shows the effect of increasing the velocity of air in the working places. (The effective temperature of the various conditions is introduced in the table to afford a base for comparison.)

It will be seen that the improvement in the conditions is solely due to increasing the velocity of the air. Various other investigations, such as those of Haldene, Harrington, and the latest findings of the Research Laboratory, indicate that the upper limit of temperature efficiently endured with manual labor is 80 deg. effective temperature.

Up to the present time considerable effort and money was spent in attempts to produce better conditions for the mine workers, and a number of failures resulted from inability to control the cooling power of the air underground.

The principal object to be attained in mine ventilation is to reduce the temperature at the working places underground, and provide a supply of fresh air to dilute and remove noxious fumes, dust, and inflammable gases resulting from the use of explosives.

The atmospheric conditions at the working places of a mine depend largely upon rock temperature, depth of the mine, and presence of water in the shafts and drifts available for evaporation. In comparatively shallow mines, and those ventilated by natural circulation, the temperature of the air in the workings depends to a greater extent upon the outside air temperature.

As the air descends the downcast shaft, its temperature is increased through the absorption of heat from the surrounding rock, and also through the higher barometric pressure. According to Haldene, the increase in rock temperature is about 1 deg. Fahr. for every 70 ft. of depth, while cases have been reported where the same increase was observed in from 40 to 180 ft. of depth. A certain quantity of heat is also added to the air at the bottom of the mine through the use of machinery, explosives and various other operations, these further raising its temperature. The moisture content of the air, as it travels down the shaft, also increases depending upon the temperature and amount of water available for evaporation.

The difficulties in maintaining proper working conditions in mines are obvious. If the incoming air is cooled by refrigeration, the greater difference in temperature between rock and air will effect a greater heat absorption, with the ultimate result of the air reaching the bottom at practically the same temperature as it would without cooling. Actual observations by Davies⁸ in Morro Velho Mine in Brazil show that the incoming air at a temperature of 76 deg. arrived at the bottom with a temperature of 101 deg., and when cooled by refrigeration to 42 deg. reached the same level at a temperature of 97.4 deg.

This proves that owing to the extremely low efficiency the use of refrigeration in mine ventilation is prohibitive. In addition, its first cost and cost of operation is many times greater than the corresponding cost of the simple and efficient equip-

⁷ Some Kata-Thermometer Observations, in Tonapah Mines, Nevada, by R. S. Lewis, *Engineering and Mining Journal Press*, March, 1924, p. 364.

⁸ The Air Cooling Plant, St. John del Rey Mining Co., Ltd., Brazil, by E. Davis, *Transactions, Institute Mining Engineering*, Vol. LXIII, London, 1922. Also, Cooling of Mine Air, U. S. Bureau of Mines, Report of Investigations 2554, by T. T. Reed and F. C. Houghten.

ment involved in the application of the new principles of air conditioning to mine ventilation.

As mentioned before, evaporation of water effects a marked lowering in the temperature of the air, especially when the latter is comparatively dry. It has been observed in a few mines that under favorable conditions the cooling resulting

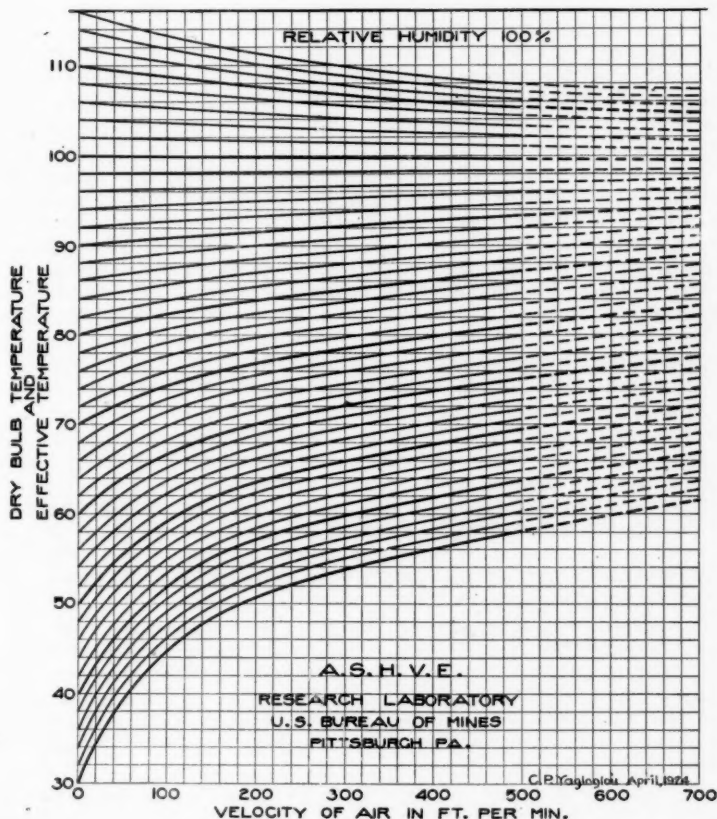


FIG. 8. VARIATION OF EFFECTIVE TEMPERATURE AND COOLING PRODUCED BY AIR MOVEMENT ACCORDING TO THE VELOCITY OF AIR, FOR CONDITIONS WITH 100% RELATIVE HUMIDITY

from evaporation of water, dripping along the walls of the downcast shaft and drifts more than counteracted the increase in temperature, and the air reached the bottom at a lower dry bulb than on entering the shaft at the top of the mine.

This principle of cooling should be frequently made use of in mines, either naturally or artificially by means of humidifiers or sprays, especially when the tem-

perature conditions are such that no reasonable velocity of the air is sufficient to produce a decided improvement.

To illustrate, the case of a hot metal mine is taken when the dry-bulb temperature was about 105 deg. and the wet bulb 88 deg. This condition corresponds to 91.7 deg. effective temperature, and the men could only remain at work for about an hour without rest. From Table I it is found that the velocity necessary to reduce this condition to 80 deg. effective temperature is above 700 ft. and therefore impracticable. If, however, the air is previously cooled by evaporation of water, its temperature will be reduced to 88 deg., and the resulting cooling by saturation alone will be 3.7 deg. effective temperature. A velocity of about 600 ft. per min. now applied will probably be sufficient to reduce the condition to 80 deg. effective temperature producing a total improvement of 11.7 deg. effective temperature.

The financial aspects of cooling by saturation and air motion can be demonstrated by a study of the cooling plant at the Morro Velho Mine. The average underground condition existing prior to March, 1920, was 101.5 deg. dry bulb and 87.0 deg. wet bulb with a wet Kata cooling power of 7.7 millicalories per sq. cm. per sec. In April, 1922, an expensive refrigerating equipment was installed at the surface with a capacity of 100,600 B.t.u. per min. together with an additional 200 h.p. Sirocco blower to effect a total air supply of 60,000 cu. ft. per min. Through these changes the dry-bulb temperature was reduced from 101.5 deg. to 97.4 deg., the wet bulb from 87.0 deg. to 76.2 deg., and the wet Kata cooling power was increased from 7.7 to 20.5 millicalories per sq. cm. per sec.

Assuming that the wet Kata-thermometer can be depended upon for velocity determinations, the computed values are 220 ft. per min. prior to March, 1920, and 540 ft. after the changes were made. The original condition (101.5 deg. dry bulb, 87 deg. wet bulb, and 220 ft. per min.) corresponds to an effective temperature of 88 deg., and was reduced by refrigeration and the increased air movement to 78.7 deg., with a resulting improvement of 88.0 deg. - 78.7 deg. = 9.3 deg. effective temperature. Of this total improvement, 7.4 deg. effective temperature is apparently due to refrigeration alone, and 1.9 deg. effective temperature or 20 per cent is due to the increased velocity.

It is of great financial significance to find now that the same improvement could be theoretically obtained by saturation and the increased velocity without the use of the refrigerating equipment. By simply saturating the air both dry bulb and effective temperature are reduced to 87 deg., when a velocity of about 560 ft. per min. will further reduce the condition to 78.7 deg. effective temperature, producing a total improvement equal to that obtained by the combination of refrigeration and air motion.

An estimate of the initial cost and cost of operation of the refrigerating plant, as compared with the corresponding costs of a simple humidifier, will disclose the enormous saving in investment and power consumption accomplished by this method.

Application to Steel Mills and Allied Industries

In the very hot industries of steel, iron, and tin manufacture, the workers depend chiefly on evaporation as the only means of eliminating bodily heat. The work is hard and is carried out under trying conditions of temperature. The men stream with perspiration and drink many kinds of liquids to replace the loss by evaporation. Owing to the heat and excessive perspiration they usually work with the upper

half of the body uncovered, and alternate short periods of work with resting periods in which they cool off.

Steel production is the most strenuous occupation in which thousands of men are engaged, and often requires exposure to high temperatures which may reach to 220 deg. fahr. a few feet from the furnace. In tin-plate rolling mills the rollerman and behinder usually stand in temperature of from 100 to 120 deg. fahr., working laboriously in the face of radiant heat from the furnaces and plates. In glass works the temperature in the vicinity of the furnace often reaches 140 deg. fahr., and the workers, like those in the metal industries, frequently rest while others take their places.

Continual exposure to these conditions is found to have an immense economic effect on the efficiency and health of the workers. They become susceptible to disease, and invariably suffer from anemia and muscular and joint pains, which eventually induce premature old age.

A great deal has been done lately in an effort to improve working conditions. Forced air blasts have been introduced in a few cases blowing fresh cold air over the heads of the men with beneficial results. Such a system lowered greatly the temperature and improved considerably the efficiency in a tube plant in Pittsburgh. Its use has also been effectual in overcoming adverse heat conditions in bottle works and tin-plate factories and thus has increased considerably the output and decreased respiratory diseases.

Besides the lowering in temperature effected by the circulation of fresh cool air, the movement of the latter greatly increases its cooling effect upon the bodies of the men, and the work proceeds with shorter periods of rest which otherwise were spent in cooling off.

The experimental evidence in hand of the cooling laws of the human body is of great value in predicting just what is expected of a certain air velocity at a given temperature and moisture content when directed upon the body of lightly clothed individuals. This information further indicates that the most efficient system of ventilating hot workshops is one in which the air is cooled by saturation, and then blown through overhead ducts directly over the workers. In addition to increasing the cooling power of the moving air, saturation provides moisture to the relatively dry air, and thus eliminates the burning effect of the dry hot air and its influence upon the respiratory organs.

A condition of 112 deg. dry bulb and 86 deg. wet bulb existed in a steel mill in the Pittsburgh district, with an average velocity of 100 ft. per min. created by the natural circulation of air. It was observed that in the hot summer months, the work did not proceed as fast as it did in winter, and improvements were contemplated through the installation of large blowers to effect a rapid circulation of air.

In summer, however, the temperature conditions outside are too high to effect any appreciable lowering in the temperature at the working places by this method. Furthermore, as mentioned previously, the effect of wind at these high temperatures is rather small to bring about a decisive improvement.

The effective temperature of the condition is 91.0 deg., and by saturating the air at 86.0 deg. and blowing it on the workers with a velocity of 400 ft. per min., it is theoretically reduced to 80.0 deg. which is a good working temperature for steel mills. The cooling produced is 91.0 deg. - 80.0 deg. = 11.0 deg. effective temperature, while if the velocity of the air is increased to 700 ft., the corresponding effective temperature will be 74.5 deg. and the resulting cooling 16.5 deg. effective temperature.

If the workers could possibly derive the full benefit of the cool air bath, these values will be realized in actual practice. In any case, the improvement will depend on the effectiveness of directing the current of air upon the workers and the temperature and humidity of the air as it strikes the bodies of the latter.

Acknowledgment

The writers are indebted to W. H. Carrier and F. C. Houghten for their interest in the work and valuable suggestions offered.

No. 707

THE HEAT GIVEN UP BY THE HUMAN BODY AND ITS EFFECT ON HEATING AND VENTILATING PROBLEMS

By C. P. YAGLOGLOU,¹ PITTSBURGH, PA.

MEMBER

Body heat being largely a problem of medical interests, has received considerable study by various physicians and others both in this country and abroad who were chiefly interested in the chemical changes that take place within the body, in order to formulate standards of nutrition requirements for various classes of workers. The results of these studies, although valuable, are of little consequence to the heating and ventilating engineer in the form available, and the purpose of this paper is to analyze the existing data from a technical view point and present it, together with supplementary results obtained in the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, in a form convenient for use and practical application in the field of heating and ventilation.

THE human body in its thermodynamic functions resembles a combined boiler and engine unit. Combustible material, in the form of food, is taken in at regular intervals and undergoes chemical decompositions, largely oxidative in nature, as a result of which heat is evolved, quantitatively equal to that developed when the same food substances are burned in a furnace. Part of this heat energy is required for the maintenance of the internal functions of the body, such as blood circulation, breathing, and other activities of the internal organs; while the remaining part is consumed in the performance of the various external physical activities observed in life.

During this continuous process of transformation, the body is capable of regulating the heat production according to the circumstantial requirement through the action of its thermostatic control. Under normal conditions the temperature of the body is considerably higher than that of the atmospheric environment, and it there loses heat to the surrounding air and objects. The constant temperature of the body indicates that the rate of heat loss is regulated so as to be equal to the rate at which it is produced.

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Heat loss from the human body under normal conditions and at rest takes place in the following different ways and approximate proportions:

	Per Cent
Radiation conduction and convection.....	73.0
Evaporation of moisture from skin.....	14.5
Evaporation of moisture from lungs.....	7.2
Warming the inspired air.....	3.5
Warming the food ingested.....	1.8

The loss of heat is largely regulated by the amount given off by radiation and convection, and evaporation of moisture from the surface, through the remarkable action of the body thermostatic temperature control. As long as the body temperature remains constant the total heat loss must equal the amount produced. To determine the heat developed within the body, or that lost, methods of direct and indirect calorimetry are used. Usually, the heat value of the food taken in, and of the oxygen absorbed in respiration, corrected for the unused heat units contained in the discharged matter, is compared with the contemporaneous heat actually lost through the various ways given above.

One of the most completely equipped and accurate calorimeters is the Atwater-Rosa-Benedict respiration calorimeter,² designed for determining the respiratory exchange and the simultaneous measurement of the quantity of heat given up by the human body. The apparatus resembles an ordinary living room furnished with the primary necessities of life, and well insulated and equipped with automatic temperature control. Chemical absorption permits direct estimation of the carbon dioxide and moisture given off in the process of respiration, and also of the oxygen absorbed, from which the amount of heat produced in the body can be calculated. The heat given up by the body is absorbed by a current of cold water flowing through copper tubes suspended from the upper wall of the chamber. In the majority of cases, results obtained by the two separate methods agree very closely.

The heat production and loss vary through wide limits according to the food taken in, clothing worn, temperature of the environment, and degree of muscular activity. It is minimum during sleeping hours, when the general system of the body is more or less inactive. This minimum value is known as the basal requirement, or basal metabolism, and represents the energy expended for the performance of the work of circulation, respiration, and vital activities of the living cells.

Table I gives the minimum heat production and loss with regards to age and sex. The values represent Aub and Du Bois' standards of basal metabolism converted to B.t.u. per sq. ft. of body surface per hour. The surface area is generally determined from the weight and height by means of Du Bois standard formula or chart,⁴ and the former is given by the equation below, expressed in British units.

$$A = W^{0.425} \times H^{0.725} \times 0.10862.$$

where A = Body surface in square feet.

W = Weight without clothing in pounds.

H = Height in inches.

For practical use, Fig. 1 has been prepared from which the surface area of the body can be readily determined from the weight and height.

² Lusk, Graham "Archives of Internal Medicine," 1915, XV, p. 793.

³ Aub and Du Bois "Archives of Internal Medicine," 1917, XIX, p. 831.

⁴ Du Bois D., and E. F., "Archives of Internal Medicine," 1916, XVII, p. 836.

TABLE 1. MINIMUM HEAT PRODUCTION AND LOSS

Age in Years	B.t.u. per Sq. Ft. per Hr.	
	Males	Females
14-16	17.0	15.9
16-18	15.9	14.8
18-20	15.2	14.0
20-30	14.7	13.7
30-40	14.6	13.5
40-50	14.2	13.3
50-60	13.8	12.9
60-70	13.5	12.6
70-80	13.1	12.2

Under ordinary conditions during waking hours and at rest, there is always a greater or less amount of muscular activity, involving an increase of heat produc-

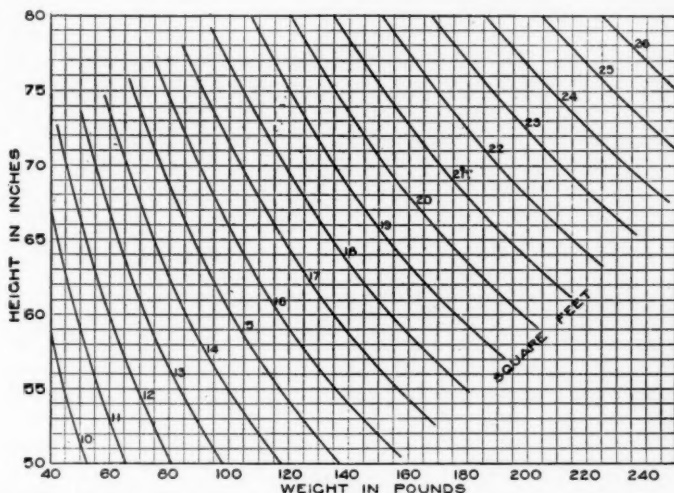


FIG. 1. CHART GIVING THE SURFACE AREA OF MEN ACCORDING TO HEIGHT AND WEIGHT

tion and loss over that of the minimum value. Ingestion of food and sitting position add appreciably to the energy requirement, so that a person, although at complete rest, loses a greater amount of heat during waking hours than while asleep. Lusk⁵ gives 278 B.t.u. for basal and 306 B.t.u. per hr. for resting heat loss in a man of 156 lb. weight and 5 ft. 7 in. tall. The corresponding values per sq. ft. of body surface are 14.1 and 15.6 B.t.u. per hr., an increase of about 11 per cent over the basal requirement.

As mentioned before, the body temperature is maintained practically constant, under different conditions of external temperature by the thermostatic mechanism in the body, which controls the production and loss of heat. The greatest regulation is done on the heat loss side, principally controlled by the amount given off by radiation, by convection, and by evaporation of moisture from the surface of the body. The relative loss by these means will naturally depend upon the temperature difference between the body and surrounding air and objects, the humidity

⁵ Lusk, Graham, "Food in War Time," Saunders & Co., 1918.

and the velocity of air. At low temperatures there is relatively little heat loss by evaporation, while at body temperature there is no heat loss by radiation and convection, all the heat produced being lost by evaporation of water from the surface. With saturated air at body temperature, heat loss becomes impossible, as a result of which the temperature of the body rises.

The total heat loss per square foot of body surface per hour for normally clothed individuals at rest and exposed to different temperature conditions with still air and moving air is shown in Fig. 2, plotted against effective temperature.⁶ The data is based on the results of a number of investigators mentioned in the figure

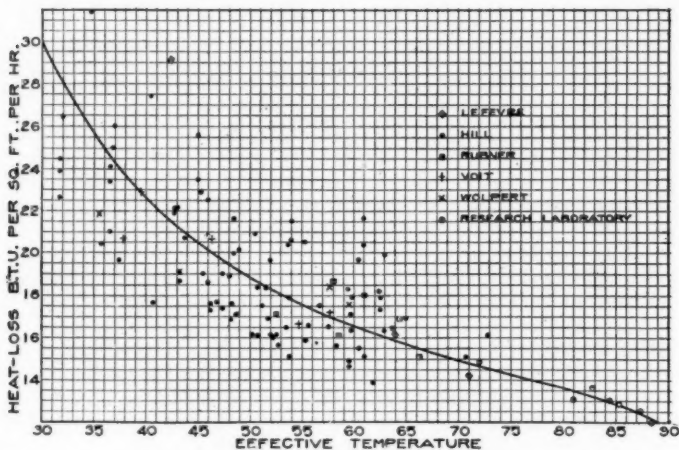


FIG. 2. CHART GIVING THE HEAT LOSS FROM THE HUMAN BODY IN B.T.U. PER SQUARE FOOT OF BODY SURFACE PER HOUR FOR VARIOUS EFFECTIVE TEMPERATURES

and is supplemented with results obtained in the psychrometric rooms of the Research Laboratory in cooperation with the U. S. Bureau of Mines and the U. S. Public Health Service.⁷

It will be observed that the heat loss is not directly proportional to effective temperature but increases at a greater rate as the temperature falls. At 30 deg. it is twice as great as at 70 deg. effective temperature, while at the normal temperature of 65 deg. effective temperature, the average individual loses heat at the rate of 15.7 B.t.u. per sq. ft. of surface per hour. For an average man of 20 sq. ft. of surface the loss is about 315 B.t.u. per hr. Of this heat 73 per cent, or 230 B.t.u. is lost by radiation and convection, and the remaining 85 B.t.u. are lost by evaporation of moisture from lungs and skin and other minor means.

Above 65 deg. effective temperature the loss is practically directly proportional to effective temperature up to 85 deg., above which a falling off is apparent. This latter part of the curve is entirely based on the results of the Research Laboratory.

⁶ Determining Equal Comfort Lines, by F. C. Houghten and C. P. Yagloglou, *JOURNAL, A.S.H. & V.E.*, March 1923; and Effective Temperature Applied to Industrial Ventilation Problems, by C. P. Yagloglou and W. E. Miller, *JOURNAL, A.S.H. & V.E.*, July 1924.

⁷ Some Physiological Reactions to High Temperatures and Humidities, by W. J. McConnell and F. C. Houghten, *JOURNAL A.S.H. & V.E.*, March 1924, pp. 141-144.

The heat production was determined from the carbon dioxide output and oxygen absorbed, from which an estimate was made of the heat loss by subtracting the heat retained in the body through a rise in temperature. Thus, the heat loss at these comparatively high temperatures is no longer equal to that produced, and only a fairly close estimation can be obtained in the amount of the former, provided the rise in body temperature does not exceed 0.5 deg. per hr. It is reasonable to assume that the heat loss will become zero at an effective temperature equal to body temperature, which, if surpassed, the body will gain heat from the surrounding air and objects by radiation, convection, and condensation of moisture on its surfaces.

A discussion of heat loss would be incomplete without considering the amount of heat produced at these higher temperatures. Fig. 3 shows a duplicate portion of Fig. 2 up to 65 deg. effective temperature, supplemented with the high temperature results of the Research Laboratory.⁸ Contrary to expectations, the rate at which heat is generated is much greater at high temperatures than at low. There is a zone of minimum heat production, between 72 deg. and 80 deg. effective temperature, above which a marked increase takes place. The body makes strenuous efforts to resist rise in temperature by promoting evaporation of water from its surface. However, there is a limit to the action of the human thermostatic control, which apparently fails above 90 deg. effective temperature.

The great variation in the data presented in Fig. 2 is probably due to the following causes:

1. Differences in the methods used in determining heat production.
2. Individual differences in the subjects employed, and clothing worn.
3. Fluctuating temperature conditions during the experiments.
4. Method of determining wet bulb temperature and velocity of air.

The majority of the low temperature experiments with still and moving air were conducted outside in the open air, where the presence of sunlight and irregularity in the magnitude of the wind undoubtedly had some influence on the results obtained.

A number of investigators believe in the constancy of heat production and loss, irrespective of moderate changes in the atmospheric environment, and little or no importance is attached by them to temperature measurements. In regard to humidity, Hill⁹ believes that changes at ordinary room temperatures are immaterial, at least as far as heat loss is concerned, while Rubner states that an increase of 12.5 per cent in the humidity of the air affects heat loss to the extent of 2 deg. rise in dry-bulb temperature. The above analysis shows that both temperature and humidity, particularly the latter at higher temperatures, are influencing factors.

Considering next the effect of air movement alone, heat loss is greatly accelerated in the presence of wind which carries away the warm and saturated air entangled in the clothing.

By means of a ventilation calorimeter, Lefevre¹⁰ has determined the heat loss by radiation and convection of an individual at rest for different temperatures and velocities, with and without clothing. His results converted to B.t.u. per sq. ft. per hr. are plotted against dry bulb temperature in Fig. 4 for velocities of 200 and 700 ft. per min. Here again the heat loss by radiation and convection is

⁸ Work cited.

⁹ Hill, L., *The Science of Ventilation and Open Air Treatment*, Part I, Medical Research Committee, Special Report, Series No. 32, p. 101.

¹⁰ Lefevre, J., *Chaleur Animale et Bioenergetique*, Paris, 1911, pp. 430-433.

not directly proportional to the temperature. Clothing suitable for spring weather was found by Lefevre to reduce the rate of heat loss by about 40 per cent.

Inspection of Figs. 2 and 3 will show that there is a definite correlation between production and loss of heat, and effective temperature, as might be expected. To establish this fact definitely, and also to find out whether the heat produced at body temperature could be determined with any reasonable degree of accuracy from the physiological reactions measured, a series of tests was run at body temperature for two extreme humidities. Advantage was taken of the fact that with a varying saturated atmospheric condition, or its equivalent, made to follow the temperature of the body as the latter rises, there can be no heat exchange between the body and air. Therefore, all the heat developed is utilized in raising the temperature of the body.

Knowing the weight of the body, the rate of rise in its temperature during exposure, and the composite specific heat of the body, the heat retained within it per hour can be easily calculated. The specific heat of the entire body is usually taken as 0.83, so that for a man weighing 150 lb., the heat retained through a rise in temperature rate of 1 the at deg. per hr. will be $150 \times 0.83 \times 1 = 124.5$ B.t.u. per hr.

Samples of the data obtained in the tests are shown in Tables 2 and 3 for 18 per cent and 100 per cent relative humidity, respectively. The variation in the temperature conditions during the tests is given in columns 2, 3, and 4, the effective temperature in the latter column following as closely as possible the rectal temperature of the subject in column 5. Observations of rectal temperature, pulse rate, and body weight for the various subjects are given in the remaining columns for every 10 minute intervals of exposure. The results of the experiments, computed in the above manner are presented in tables 4 and 5. The heat retained in the body per hour is calculated separately for each subject and given in column 10. This divided by the sq. ft. of skin surface (column 11) gives the heat per square foot of surface per hour. It will be observed in column 12 that this latter quantity varies to a certain extent with the different individuals, as is to be expected, but the average is about the same for all tests. The average of the low humidity tests is 16.5 B.t.u. per sq. ft. of surface per hour, and that for the saturated tests is also 16.5 B.t.u., an exceptionally close agreement, showing the constancy in the thermal changes of the body with constant effective temperature.

As far as the quantitative results are concerned, the value of 16.5 B.t.u. per hr. seems to be low when compared with the heat production of 28 B.t.u. per sq. ft. per hr. at the average temperature of 101 deg. effective temperature, as determined from the respiratory exchange in Fig. 3. It is possible that the heat is not always distributed uniformly throughout the body for the rise in rectal temperature to represent the average rise in the temperature of the entire body. Calorimetric observations previously made by a number of investigators have indicated this condition. When there was a wide variation in rectal temperature during the experiments, direct and indirect calorimetry did not check as closely as when the variation was small. However, the results are valuable as they prove indirectly, yet conclusively, that the heat production is constant for constant effective temperature regardless of temperature or humidity alone. Since the heat production under normal circumstances equals the heat loss the effective temperature lines are also lines of equal heat loss. This follows also from a consideration of other physiological reactions which are constant for constant effective temperature.

By far the most powerful factor in the body's heat balance is muscular activity. Heat production during work is greatly increased due to the increased activity of the muscles. All of this heat is not, however, eliminated from the surface of the

TABLE 4. DATA AND RESULTS OF LOW HUMIDITY TESTS
Range in Test Conditions

Test No.	Dry Bulb	Wet Bulb	Relative Humidity	Effective Temperature	Rise in Rectal Temperature, °F per Hr.	Increase in Pulse Rate	Body Weight Lbs.	Heat Retained in Body, B.t.u. per Hr.	Body Surface Sq. Ft.	B.t.u. per Sq. Ft. per Hr.	Average B.t.u. per Sq. Ft. per Hr.
1	140.2 to 145.3	91.8 to 95.0	18%	99.6-102.0	2.1	49	152.8	266.3	21.08	12.6	15.9
					2.9	59	134.1	322.8	18.49	17.5	
					2.8	86	130.7	303.7	18.40	16.5	
					2.8	52	129.6	301.2	17.88	16.8	
2	147.2 to 155.3	89.4 to 93.0	11%	99.2-102.0	2.8	48	152.9	355.3	21.08	16.8	17.1
					2.7	56	133.4	299.0	18.45	16.2	
					2.6	86	130.5	281.6	18.39	15.3	
					3.3	72	129.8	355.5	17.89	19.9	

TABLE 5. DATA AND RESULTS OF SATURATED TESTS

Test No.	Dry Bulb	Wet Bulb	Relative Humidity	Effective Temperature	Rise in Rectal Temperature, °F per Hr.	Increase in Pulse Rate	Body Weight Lbs.	Heat Retained in Body, B.t.u. per Hr.	Body Surface Sq. Ft.	B.t.u. per Sq. Ft. per Hr.	Average B.t.u. per Sq. Ft. per Hr.
3	99.9 to 102.8	99.9 to 102.7	100%	99.9-102.8	2.6	68	142.0	306.4	19.20	16.0	16.6
					2.6	62	134.7	290.7	18.53	15.7	
					2.8	67	131.0	304.4	18.42	16.5	
					3.0	74	129.5	322.5	17.87	18.0	
4	98.8 to 101.9	99.7 to 101.9	100%	98.8-101.9	2.6	66	141.0	304.3	19.15	15.9	16.4
					3.0	52	134.6	335.2	18.52	18.1	
					2.6	80	132.0	284.9	18.46	15.4	
					2.7	54	128.4	287.7	17.64	16.3	
					2.7	68	129.8	290.9	17.89	16.3	

body. A certain amount appears as external mechanical work, so that to obtain the quantity given off in the form of heat the mechanical equivalent of the external work performed is subtracted from the heat generated.

Table 6 is based on the calorimetric observations of Atwater and Benedict,¹¹ and shows the average heat loss by various means in a large number of experiments with several individuals both at rest and doing measured amounts of work on a stationary bicycle. The heat loss by radiation and convection during work increased 137 per cent over that for the men at rest, the heat loss by evaporation increased 85 per cent, while the total heat production increased by 177 per cent.

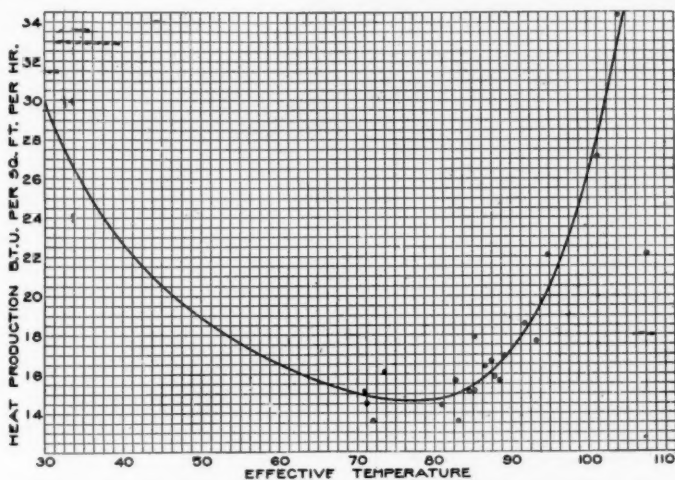


FIG. 3. CHART GIVING THE HEAT PRODUCED IN THE HUMAN BODY IN B.T.U. PER SQUARE FOOT OF BODY SURFACE PER HOUR FOR VARIOUS EFFECTIVE TEMPERATURES

TABLE 6. HEAT ACCOUNT AT REST AND DURING WORK

Avenues of Heat Loss	B.T.U. per Hour		Increase per cent
	At rest	At work	
Radiation and convection.....	311	736	137
Evaporation from lungs and skin.....	102	189	85
Urine and feces.....	5	5	0
Equivalent of mechanical work.....		228	
Total.....	418	1158	177

The heat production of men in work of various rates of output has been studied by a number of investigators, principally by Benedict and Cathcart,¹² and Benedict and Carpenter,¹³ on bicycle ergometers. From their observations the heat loss per square foot of body surface has been calculated by subtracting the mechanical equivalent of the external work done. The results are shown in Fig. 5 plotted against rate of work in foot pounds per hour. Apparently heat loss is not directly proportional to the output due to the variation in the efficiency of the human body

¹¹ Atwater and Benedict, U. S. Dept. of Agriculture, Office Experiment Station, Bulletin No. 136 1903.

¹² Benedict & Cathcart, Muscular Work, Carnegie Inst. Wash., 1913.

¹³ Benedict and Carpenter, U. S. Dept. of Agriculture, Office Experiment Station, Bulletin No. 208, 1909.

in transforming heat into mechanical work. The mechanical efficiency of the human body is the ratio of the amount of work done, expressed in heat units, to the increase in heat production due to work, above the resting value, during the same period of time. The average efficiency is about 20 per cent. In other words only one-fifth of the total increase in the heat production due to work is expended in useful work and the remaining amount is given off as body heat. As a matter of fact, this heat is not eliminated as fast as it is produced, unless the temperature of the environment is low enough to effect this loss. Part of it is retained in the body, giving rise to physiological reactions as represented by increase in body temperature, pulse rate, and rate of respiration, until the body temperature rises to a degree at which a balance is reached between the rate of heat production and loss, de-

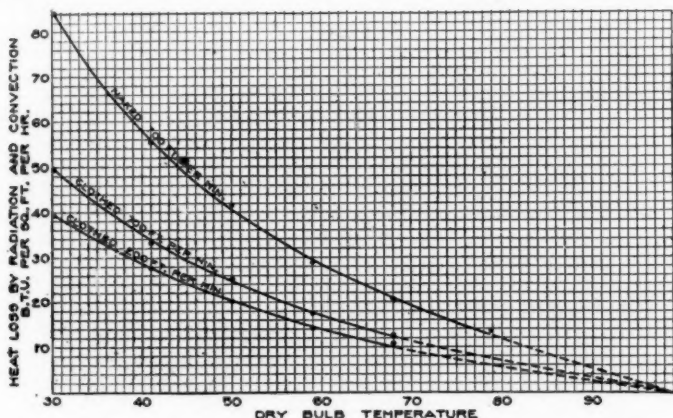


FIG. 4. CHART GIVING THE HEAT LOSS BY RADIATION AND CONVECTION FROM THE HUMAN BODY IN B.T.U. PER SQUARE FOOT OF BODY SURFACE PER HOUR FOR VARIOUS DRY BULB TEMPERATURES

pending on the external temperature. For the higher outputs this balance is not attained within physiological measures compatible with life, and intermittent periods of rest are necessary for cooling off. In this respect the heating and ventilating engineer can do a great deal towards providing proper cooling power in the air to secure the greatest efficiency and output with the least amount of fatigue.

As to the effect which external temperature conditions might have on the heat production during work, Rubner found that neither temperature nor clothing affects the amount produced. They influence only the quantity of perspiration available for evaporation through which the body makes an effort to maintain its normal temperature by physical regulation. It follows, therefore, that the heat to be eliminated from the surface of the body (not the heat loss) as given in Fig. 5 remains constant, regardless of external temperature provided that sufficient mechanical work is done to maintain the temperature of the body at low temperature conditions.

From observations of Becker and Hämäläinen¹⁴ and deductions made by Lusk,

¹⁴ Becker and Hämäläinen, *Scandinavisches Archiv. für Physiologie*, 1914, XXXI, p. 198, cited after L. Hill, *The Science of Ventilation and Open Air Treatment*, Part I, Medical Research Council London, 1919, p. 54.

Hill, and Greenwood, Table 7 has been prepared, showing the heat outputs for men at various trades. In computing these values the mechanical efficiency was taken as 20 per cent and the resting heat loss 315 B.t.u. per hr. for the average body surface of 20 sq. ft.

The class of brain workers (not given in the table) loses only a little more than the resting man, averaging about 360 B.t.u. per hr., while lumbermen and other out-of-doors laborers, who in addition to hard work often face extremely cold weather, lose 1510 B.t.u. per hr., the highest value in the trades. It remains to be found by experiment, what will be the optimum temperature for each representative type

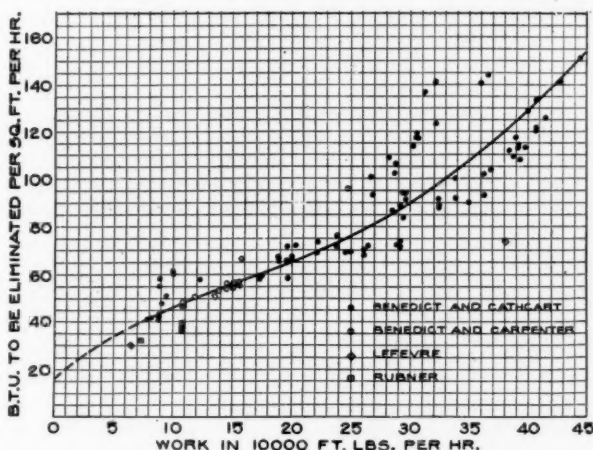


FIG. 5. CHART GIVING HEAT TO BE ELIMINATED FROM THE HUMAN BODY IN B.T.U. PER SQUARE FOOT OF BODY SURFACE PER HOUR FOR VARIOUS RATES OF WORK

of worker at which heat loss will take place without undue rise in temperature and with the least amount of discomfort.

TABLE 7. HEAT OUTPUT DURING WORK FOR MEN AT TRADES

Occupation	Total Heat To Be Eliminated. B.t.u. per Hr.
Tailor.....	440
Bookbinder.....	545
Shoemaker.....	574
Carpenter.....	646-783
Metal Worker.....	718
Painter.....	725
Stonemason.....	1172
Man sawing wood.....	1395

Adequate ventilation in crowded places, such as theaters, auditoriums and schoolrooms, demands chiefly the prevention of stagnation of body heat. Every adult occupant constitutes a stove giving off heat at the average rate of 350 B.t.u. per hr., in this sedentary condition, which must be removed if the place is to be kept at a comfortable temperature. Of this total heat 240 B.t.u. are lost per person per hour by radiation, convection, and heating the inspired air. Each person

gives off enough heat to raise the temperature of the 1800 cu. ft. of air per hr. required by ventilation laws through

$$\frac{240 \times 55}{1800} = 7.9 \text{ deg.}$$

(55 = cubic feet of air heated through 1 deg. by 1 B.t.u. at 70 deg.)

The moisture given off from the lungs and skin per person per hour under these conditions is about 0.106 lb., so that the moisture content of the ventilating current increases from 53 grains per lb. (at 70 deg. dry bulb and 58 deg. wet bulb) to

$$53 + \frac{0.106 \times 7000}{1800 \times 0.075} = 59 \text{ grains per lb.}$$

(0.075 = density of air at 70 deg.).

Referring now to a psychrometric chart it is found that the wet-bulb temperature of the ventilating current has increased from 58 deg. to 62 deg. and its effective temperature from 64.5 deg. to 69.5 deg., due solely to the heat and moisture absorbed from the bodies of the occupants, not taking into consideration the heat given off by lights and equipment, nor the heat lost through the walls and roof of the building.

The New York State Commission on Ventilation bases the air supply required in schools on the heat given off by the pupils, estimated at 300 B.t.u. per pupil per hour sitting in the standard American class room. According to the calculations of the Commission, 1800 cu. ft. of air is needed per pupil per hour, which properly introduced and diffused throughout the room, will absorb and carry off the heat and moisture given up by the occupants, and provide sufficient ventilation.

In factories and workshops a considerably greater supply of air is needed to remove the large amount of heat developed by the men at work, in addition to that evolved by the machinery in operation, and other sources incidental to the processes of manufacture. From Fig. 5 and Table 6 an estimate could be made of the quantity of heat given off from the bodies of the workers. The average man doing physical labor performs 75,000 ft. lb. per hr. At this output under normal working conditions he loses 40 B.t.u. per sq. ft. per hr., or a total of 800 B.t.u. for a body surface of 20 sq. ft. From this it is clear that a very large volume of air is needed to keep down the temperature. To this end, a method of conditioning the air has been proposed by the Research Laboratory¹⁸ in which the air is cooled by passing it through a humidifier, and then blown through overhead ducts directly upon the workers. This arrangement provides maximum cooling with a minimum supply of air.

References.—HILL, L. E., (1919) *The Science of Ventilation and Open Air Treatment*, Part I; Medical Research Committee, Special Report Series No. 32, pp. 41-164; (1920) *Ibid.*, Part II; (1923) *The Kata Thermometer in Studies of Body Heat and Efficiency*, Medical Research Committee, Special Report Series No. 73, pp. 144-187; HILL, L. E. and CAMPBELL, J. A., (1922) *Observations on Metabolism*, *The Lancet*, Vol. I; LEFEVRE, J., (1911) *Chaleur Animale et Bioenergetique*, Paris, pp. 512-519; LUSK, G., (1919) *The Elements of the Science of Nutrition*, pp. 1-151 and 309-334; VOIT, (Series publications 1860-1902) cited after Lusk, *Ibid.*; RUBNER, M., (Series publications 1879-1914) cited after Lusk, *Ibid.*; WOLPERT, *Archives für Hygiene*, 1898, XXXIII, 206, 1901, XXXIV, 298, and cited after Hill, L. E. (1919); WALKER, I. H., (1907) *Metabolism and Practical Medicine*; ANSEUX, G., (1890) *L'influence de la température extérieure sur la production de chaleur chez les animaux à sang chaud*, *Recherches de calorimétrie*, *Bulletins L'Académie Royale Belgique*, 3me série, 20, Bruxelles, pp. 599-614; MAGNÉ, H., (1920) *Influence de la température extérieure sur la grandeur de la dépense d'énergie occasionnée par le travail musculaire*, *Comptes Rendus des Sciences et Mémoires de la Société de Biologie*, 83, pp. 396-397; CAMPBELL, J. A., D. HARWOOD-ASH and HILL, L., *The Effect of Cooling Power of the Atmosphere on Body Metabolism*, *Journal of Physiology*, Vol. LV, 1921, pp. 259-264.

¹⁸ *Effective Temperature Applied to Industrial Ventilation Problems*, by C. P. Yagloglou and W. E. Miller, *JOURNAL A.S.H.V.E.*, July 1924.

No. 708

MODERN TREND IN THE SCIENCE OF VENTILATION

By PERRY WEST,¹ NEWARK, N. J.

MEMBER

Two Decades of Evolution in Ventilation

THE science of ventilation began its existence with, and is still growing out of the fact that wherever human beings assemble within an enclosed space the atmosphere within this space will become vitiated unless proper provisions are made to prevent it. It is the difficulty of determining and applying this prevention that has kept physiologists and ventilating engineers busy for so many years endeavoring to produce something like satisfactory results. The commonly accepted usage of the term *vitiated atmosphere* has long been the designation of those conditions which cause unpleasant, uncomfortable or unhealthful physiological reactions, but our interpretations of the true causes and meanings of these reactions have undergone many radical changes with the progress of the art and our growth in its knowledge.

In other words the physiological effects of poor ventilation have continued to manifest themselves in much the same way but our knowledge of these manifestations are continually changing. Among the effects, which have received the greatest amount of study and which are now generally recognized as direct results of poor ventilation, are the following: drowsiness, headache, loss of physical vitality, feeling of suffocation, temperature discomfort, brain fatigue, irritation of the membranes of throat, nose and lungs, infection, drying and cracking of and the causing of unnatural discharges from these membranes, disagreeable odors, loss of appetite, nervousness and general nausea.

It is only within the last 20 years that all of these manifestations of poor ventilation have been definitely recognized and the present era of ventilation started.

Prior to this time practically all artificial ventilation was attempted on what might be termed a quantity basis, based on the theory that the carbon dioxide exhaled by persons in an occupied space was the primary cause of such manifestations as were then recognized. As a result of this theory it was believed that the most important factor in ventilation was the quantity of air required from outside in order to maintain an atmosphere containing not more than 10 parts of carbon dioxide per 10,000 parts by volume, within any given space. This

¹ Consulting Engineer.

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theory of artificial ventilation has long been deposed and discredited among engineers, scientists and physiologists, but there are those who still cling to the theory, either in its original form, or in one of its many modifications.

When it was first learned that the quantity of carbon dioxide ordinarily found in even poorly ventilated spaces could not, of itself, be entirely responsible for the unsatisfactory conditions encountered under such circumstances, the possibility of other causes began to be seriously investigated.

There followed a period in which it was believed that some form of poisonous effluvium was exhaled with the human breath was responsible for the vitiation of the atmosphere within spaces occupied by human beings. Later this notion was disproved and it was believed that something might be excreted from the pores of the skin, or that small particles might be given off from the body or the internal membranes, so that the atmosphere became thus contaminated with matter which upon decomposition formed toxins or poisons to cause the effects noted. This rather fanciful theory was soon discredited, however, and then began the real study of the subject which has finally brought us to the conclusions upon which we are now working.

The basic idea of ventilation today is quality rather than quantity, or the proper conditioning and distributing of a small quantity of air by efficient compact means rather than poor conditioning and distributing of larger quantities with cumbersome apparatus too expensive to be kept in operation.

The real starting point of the modern trend of ventilation in this country can be traced to the presentation by Dr. W. A. Evans of the Chicago Health Board, and Dr. Luther H. Guillick of the Russell Sage Foundation at the 1911 Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS of a very formidable case against mechanical ventilation, as it was then conducted.

Prior to this time investigators such as Leblanc (in 1842); Claude Bernard (in 1857); Hermans (in 1883); Billings, Mitchel, and Bergey (in 1895); and Flugge (in 1905) had seriously questioned the carbon dioxide *bogy* and indicated that physical conditions of the atmosphere, like temperature and humidity, have a greater bearing upon the quality of air for ventilation than the carbon dioxide content.

It was forcibly brought to light for the first time at this 1911 Meeting that the great majority of mechanical ventilating systems installed in hospitals, schools, public buildings, etc., were not efficacious and did not produce the comfortable and healthful indoor-air conditions desired. A storm of complaints which had been gathering for years against artificial ventilation burst forth at this Meeting, and as a consequence there was considerable clearing of the atmosphere. The facts that hospital patients and anaemic school children generally showed the greatest improvements in open-air rooms, or in rooms without artificial ventilation and that operation of the elaborate and expensive ventilating plants in modern buildings was rapidly being discontinued were forcibly brought to the front.

In the discussion at this Meeting it was realized that the majority of the people in this country who were vitally interested in the problems of ventilating knew very little about proper ventilation requirements. A general confession of ignorance and confusion resulted so that enlightenment and improvement were bound to follow. One important conclusion reached at the Meeting was that while the heating and ventilating engineers had been continuously improving and refining the mechanical apparatus and were in a position to furnish most any condition of indoor atmosphere desired, the doctors and physiologists had made little or no progress in

the matter of determining just what atmospheric conditions were best suited for the maintenance of proper indoor ventilation. The engineers had previously proceeded on the theory that the supplying of a sufficient quantity of reasonably clean air from out of doors and the exhausting of a proper quantity of foul air, with reasonable control over temperature and distribution would constitute good ventilation. The doctors pointed out, however, that this old theory of ventilation, based primarily upon dilution for the purpose of keeping the carbon dioxide content down to a certain point did not produce satisfactory results. *Canned air* was the term applied to ventilation and it was found to be enervating and dileterious to the membranes of the internal air passages.

Following a period of general dissatisfaction and lack of definite data upon which to proceed with the establishment of better ventilation, there came a period of very active study and investigation by a number of doctors, physicists, and heating and ventilating engineers. Among those whose work contributed materially in this connection to the present state of the art may be mentioned: Dr. E. Vernon Hill of the Chicago Health Commission who has been connected with this development for the last 15 years, Dr. Leonard Hill of London who has been working on the subject for about the same period; Drs. Francis S. Lee, and Ernest L. Scott of the Department of Physiology of Columbia University who did notable work from 1914 to 1916; Dr. Wolff Freudenthal who has worked on the subject since 1900; Dr. James Alexander Miller who collaborated with the New York State Commission on Ventilation; Dr. Gerhard Cocks who also collaborated with the New York State Commission on Ventilation; Dr. Frederick W. Eastman of Columbia School of Medicine, G. W. Jones and W. P. Yant chemists of the Bureau of Mines, and Dr. W. J. McConnell of the United States Public Health Service who cooperated with F. C. Houghten of the Society's Research Laboratory; the late Dean John R. Allen, Dean F. Paul Anderson, Jay R. McColl, E. S. Hallet and W. H. Carrier members of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Soon after this new era of ventilation was entered upon the Society's Research Laboratory was inaugurated at the United States Bureau of Mines in Pittsburgh, and has taken a leading part in the establishment of new standards and methods of ventilation.

Some Theoretical Standards of Ventilation

The result of all of this has brought us to the present status of the art where it is no longer felt that the chemical composition of the air is the important factor but that proper ventilation depends more upon a number of other factors which may be stated in the order of their importance as follows:

1. Air supply
2. Air temperature
3. Air cleanliness in reference to its freedom from dust and other suspended matter
4. Air sanitation with reference to its freedom from bacteria.
5. Relative humidity
6. Distribution
7. Air motion
8. Freedom from odors
9. Freedom from other injurious substances
10. Freedom from monotony, with reference to noise and too much regularity of indoor conditions

Air supply is still put at the head of the list for the reason that while it is no longer considered to be the all important factor in ventilation the amount of air to be supplied per person or the number of air changes to be furnished for any particular space will always be the starting point, for without air supply there can be no artificial ventilation.

The air supply is so vitally affected by the other factors mentioned that it cannot be determined independently and it will be seen, that while this item is placed at the head of the list for the reason that it is the natural vehicle upon which the structure is carried, its importance beyond this point becomes subordinate to these other factors.

Air temperature is second for it has been proved by practically all of the accredited experimenters that overheating is more detrimental to the quality of ventilation than any other one thing.

Air cleanliness is third for it has to do with human health both from the standpoint of freedom from dust and other suspended substances which irritate and clog the air passages, and from the standpoint of freedom from bacteria and other media of infection carried along with these substances which constitute the dirt in air.

Air sanitation is fourth as it also affects human health and is correlated with the third item.

Relative humidity is fifth, not because it is of so much less importance than air supply and temperature but because it also bears such an intimate relationship with these two items that it receives a part of its due consideration in their determination. This factor will be further referred to in connection with air supply and air temperature in connection with which other factors are involved.

Distribution is sixth for a similar reason, for while it occupies a much more important place than this position might indicate, it is so intimately connected with the effective air supply that it receives a part of its consideration therewith.

Air motion is seventh in the same way, as it too receives a certain amount of its consideration in connection with effective temperature.

Freedom from odors is eighth for the reason that while odors may become quite disagreeable and even nauseating they are seldom dangerous or permanently detrimental to health.

Freedom from other injurious substances is ninth, because these substances are so seldom found in ordinary ventilating practice and must be practically eliminated in any case.

Freedom from monotony is tenth because it has to do with the last refinements and the psychology of ventilation only.

The first two steps in the new era of ventilation, first—the discovery and admission of our ignorance and second—the recognition of these important factors.

The next steps were to determine what bearing each of these factors had upon ventilation and to devise some practical means of measuring and charting these effects in comparable terms.

This work was undertaken by Dr. E. Vernon Hill, who assisted by O. W. Armspach, devised the Synthetic Air Chart which was adopted as the Society's standard in 1920.

The Synthetic Air Chart takes air supply into account under the heating of CO_2 . The scale for this factor is based on the assumption that 300 parts of CO_2 in 10,000 parts of air together with the other vitiation which would accompany this quantity of CO_2 when exhaled with the human breath might produce results that would be

permanently injurious to health. This is taken as the point where the quality of ventilation would drop to zero as far as this factor is concerned and the scale between this and the point of perfection, where no CO_2 is present, is evenly divided so that for each part of CO_2 in 10,000 (above that ordinarily contained in the outside atmosphere and assumed at 4 parts in 10,000) a deduction of $\frac{1}{3}$ per cent is made for the particular column, or department, and 0.3 per cent for the final per cent of perfection column, from 100 per cent which represents the point of perfection.

Air temperature together with air motion and relative humidity are represented on the Synthetic Air Chart under the column of wet bulb difference. These three factors are combined for the simple reason that the sensible or effective temperature depends not alone upon the dry bulb temperature but upon the relative humidity and air motion as well. Conclusions at the time this chart was devised were that the sensible temperature varied directly with the wet bulb temperature, that it dropped about 3 deg. for the first 100 ft. per min. of air motion and about 2 deg. for each 100 ft. per min. of additional air motion for an adult at rest. Also that there was a rise of about $1\frac{1}{2}$ deg. in the sensible temperature for an adult between each of the following states of activity *i. e.*, at rest, light work, moderate work and hard work.

The scale for these factors is based on the assumption that a wet bulb temperature of 106 deg., without air motion, would soon cause permanent injury to health and this is taken as the point where the quality of ventilation for these factors would drop to zero.

Fifty-six degrees wet bulb temperature, with 70 deg. dry bulb temperature was taken as the optimum point of human health and comfort without air motion and was assumed to represent 100 per cent quality. The scale is evenly divided between these two points so that each degree of difference between the observed wet bulb temperature (after correction for air motion and condition of activity) and the ideal of 56 deg. represents a deduction of 2 per cent to be made from the 100 per cent for this column. This is taken to be equivalent to a deduction of 1.8 per cent per degree difference in the final percentage of perfection column.

Within the past two years these determinations have been somewhat modified by the findings of the Society's Research Laboratory as shown in the comfort charts to be found in the Society's publications.¹

One correction is, that the sensible or effective temperature does not vary directly with the wet bulb temperature, but along lines lying about midway between the wet and dry bulb lines, within the usual temperature range for ventilation without air motion, and approaching the dry bulb lines as the air motion reaches 500 ft. per min. Another correction is that the first 100 ft. per min. of air motion produces a reduction in the sensible temperature of about 3.3 deg. instead of 3 deg., 1.7 deg. more for the next 100 ft., 1.5 deg. more for the next 100 ft., and 1 deg. for each additional 100 ft. per min. velocity up to 500 ft. per min., all based on the still air effective comfort lines' temperature of 64 deg. These new values should be substituted for wet bulb temperature differences in the Synthetic Air Chart and the entire column be headed effective temperature difference instead of wet bulb difference.

Dust, bacteria and odors are grouped under one heading in three separate columns and arranged for penalties somewhat similar to those for the two factors referred to previously. The dust column is arranged so that each 10,000 particles

¹ Cooling Effect on Human Beings Produced by Various Air Velocities, by F. C. Houghten and C. P. Yagloglou, p. 193, TRANSACTIONS, 1924. Also p. 133, GUIDE, 1923.

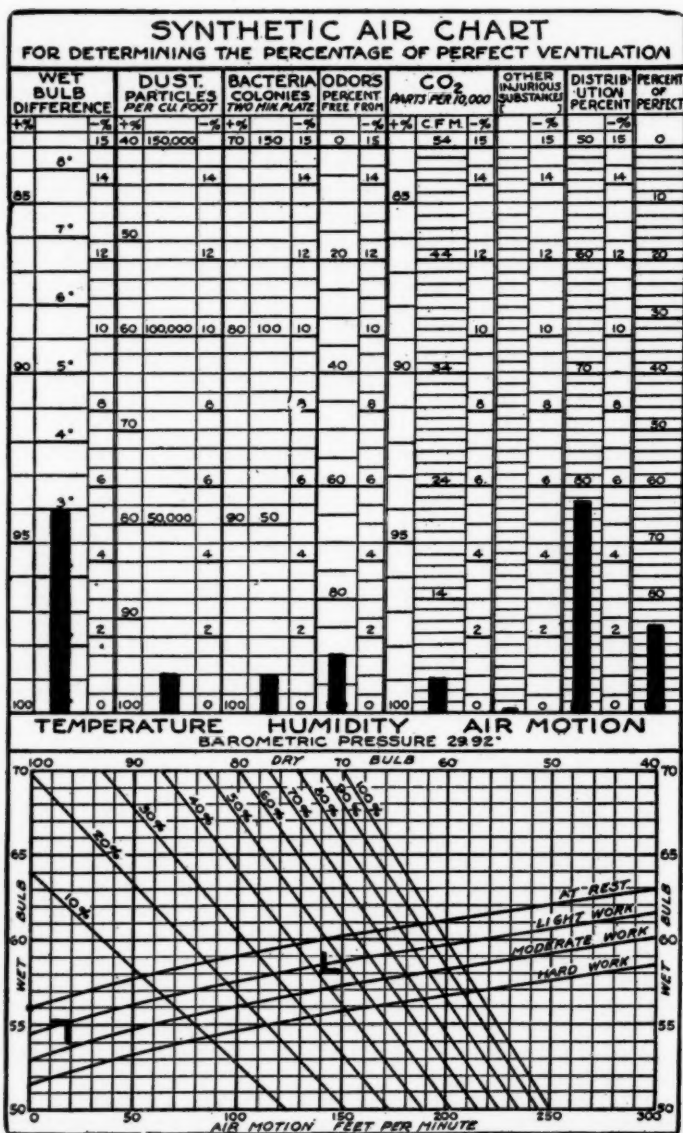


FIG. 1. THE SYNTHETIC AIR CHART

Note: For a more complete description and methods of use of the Synthetic Air Chart see the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS GUIDE, 1923, p. 123.

of dust per cu. ft. represents a penalty of 4 per cent for this particular column or department and 1 per cent in the final percentage of perfection column. The bacteria column is arranged so that each 5 colonies represent a penalty of 2 per cent for this particular column or department and 1 per cent in the final percentage of perfection column. The odors' column is arranged so that each 10 per cent below the standard of perfection of freedom from odors represents a penalty of 3 per cent in the final percentage of perfection column.

There is a separate column for distribution arranged with a scale imposing a penalty of 15 per cent for 50 per cent perfection of distribution and graduated so as to show a penalty of 1 per cent in the final percentage of perfection column for each $3\frac{1}{2}$ per cent of deficiency in the quality of distribution below perfection. Other injurious substances are treated in a separate column having an arbitrary scale.

Finally there is a percentage of perfection column where the difference between the sum of all the penalizations and 100 per cent represents the composite percentage of perfection for the ventilation conditions represented.

Taken all in all this chart is a fairly satisfactory measuring medium for quality of ventilation and covers all of the factors referred to above with the exception of *freedom from monotony*. It needs to be simplified so that it may become more generally useful and the relative values of some of the scales may require some readjustments.

The readings of all of the plus percentage columns should be reversed so as to read up from a zero or minimum base line toward the top of the chart so as to be in keeping with one's general practice in the use of charts. The deduction columns should all read downward so that the top of the $\frac{1}{8}$ in. heavy black vertical line to be charted in the center of each column will represent both the percentage of penalty and the remaining percentage of perfection for each particular factor. The CO_2 column and the distribution column might be combined into one column under the heading of effective air supply. The comfort chart should be revised in accordance with the latest authentic data on the subject from the Society's Research Laboratory.

The subject of moderate air motions, within the scope of ordinary ventilation practice, needs more investigation and the effects of the different states of activity of occupants needs to be checked by the Laboratory.

The Synthetic Air Chart was adopted five years ago, but it has seen but little practical use for two reasons. One is that it needs simplifications somewhat along the lines suggested above and the other is that it has remained as nothing more than a measuring medium by which the percentage of perfection of any existing conditions could be charted and read off or compared with any other so charted set of conditions, but there has been no key to define what percentage of ventilation, as shown by this chart, would be considered good standard practice for any particular building, class of buildings or other ventilation requirement.

The lack of this key has greatly retarded the establishment of proper ventilation standards. Fortunately, within the past year Dr. Hill has come out with such a key and it is given herewith as the first practical attempt to define what ventilation standards according to this chart should be maintained for the different requirements of spaces for which ventilation is ordinarily attempted.

Another thing which has materially retarded the establishment of better ventilation is the persistent stand which practically all of our scientists and quite a few of our engineers have maintained against the recommending of any standards cover-

E. Vernon Hill's Table of Recommended Percentages of Ventilation Perfection for Different Classes of Buildings When Tested According to the Synthetic Air Chart

<i>Schools</i>	New Buildings %	Existing Buildings %
Class Rooms	95	90
Manual Training Rooms.....	90	85
Domestic Science Rooms.....	90	85
Assembly Rooms.....	90	85
Toilet Rooms.....	85	80
Corridors.....	85	80
Churches.....	85	80
<i>Hospitals</i>		
Wards.....	95	90
Operating Rooms.....	98	93
Other Rooms.....	90	85
<i>Theatres</i>		
Seating Sections.....	90	85
Dressing Rooms, etc.....	85	80
Dance, Lodge and Assembly Halls.....	88	83
<i>Office Buildings</i>		
Offices in office buildings or other buildings where persons are continuously employed.....	90	85
<i>Factory Buildings</i>		

The percentage desirable for factory buildings will vary over a considerable range, depending upon the character of the work and of the process employed, modified to a considerable degree by the dust content of the air and the possibility of maintaining it free from objectionable dust and fumes. This will require a careful classification and considerable study.

ing the preference in methods or apparatus to be used for the production of any predetermined standard of ventilation.

When it was first realized that mechanical ventilation was not satisfactory and that open-air rooms and open-window ventilation appeared to be more satisfactory, a great controversy was started as to whether natural or mechanical ventilation was the better and as to preferable methods of mechanical ventilation. Later many of those who did not care to enter this controversial field took the position of trying to define ventilation abstractly without any reference whatever to the method or means of its production.

For a long time, therefore, and up to within the past year the scientists and physicists have maintained the position that certain standards of ventilation should be produced and that there was no practical way of specifying or even suggesting any preferable method of securing these desired results and that the only approved method of procedure was to use whatever way might be thought necessary and then if it met the standard it would be approved but if it failed to meet the standard some other method must be tried. In other words the tendency in this direction was to take a position analogous to that of a doctor who might say that he had no particular method for treating your case but would go ahead with a treatment on the basis that it would be decided from the results as to whether the treatment was correct, or of the lawyer who might say that he had no particular plan for

trying your case, but would go ahead and decide from the results as to whether the plan used was a good one.

Applied to ventilation this idea, of course, has been proven unsatisfactory since the architect, the engineer or the owner who wishes to produce a ventilating plant which will be satisfactory for any particular operation must, before plans and specifications are drawn and before the work is actually installed know what kind of an apparatus will produce the desired results. After the apparatus is designed and installed it is too late to change to some other that might have been better.

The New York State Commission on Ventilation started out in 1914 to establish proper standards of ventilation but after working for about six years did little more than confirm the already existing opinions, that overheating was one of the greatest evils of artificial ventilation, and that the carbon dioxide content of the air was not the most important index of its quality.

Fortunately Dr. Hill has come to the rescue again this year with another key to this situation and given a table showing the quality of ventilation that may be expected from the several methods of ventilation, generally employed.

This table is given below.

E. Vernon Hill's Suggested Types of Equipment for Different Synthetic Air Chart Percentages

The following classification is given to assist the engineer in selecting the type of equipment necessary to comply with certain percentages on the Synthetic Air Chart. It will be understood that considerable variation will be found in a certain class of equipment due to individual ideas on the part of the designer, the character of the installation work, the location of the building in which the equipment is installed, etc., nevertheless, it can be stated that if the equipment is properly designed, installed and operated it will give at least the percentage under the classification. The classification, moreover, is not arbitrary, but based on sound logic, amply sustained by experience.

Competent engineers at the present time do not look upon ventilation problems as dealing with the unknown or mysterious. The relations between temperature, humidity and air motion necessary for comfort have been worked out with a fair degree of accuracy. The necessity for clean air is understood and its proper distribution is only a matter of good engineering. To design a 100 per cent apparatus, therefore, it is necessary to provide apparatus for heating and humidifying the air, for thoroughly cleaning it and properly distributing it with controlling devices that will maintain the temperature and humidity in conformity with the zero equivalent temperature line as given by the Research Laboratory. Any part of the equipment that is omitted and any part that is inefficient will reduce the final percentage in a direct ratio.

Class A—100 per cent equipment. A mechanical supply and exhaust system consisting of the following:

1. Positive air supply having a maximum capacity of 30 c.f.m. per occupant
2. Mechanical exhaust equipment with exhaust registers effectively located
3. Perfect air distribution
4. Accurate automatic temperature control
5. Efficient humidifying devices
6. Accurate automatic humidity controlling apparatus
7. Efficient air washers, filters or other air-cleaning devices, having an efficiency not less than 99 per cent

It is understood that a 100 per cent efficient equipment is a physical impossibility owing to the fact that to secure a 100 per cent result would necessarily mean that air cleaning devices be 100 per cent efficient; that temperature and humidity control maintain temperature and humidity conditions absolutely on the comfort curve; that air distribution be perfect, etc. All these results cannot be obtained although a 99 per cent apparatus and an approximately 99 per cent test by the Synthetic Air Chart is possible.

Class B—95 per cent equipment. Mechanical supply system consisting of the following:

1. A positive air supply with a maximum capacity of 30 c.f.m. per occupant
2. A well designed gravity exhaust system
3. Efficient air distribution
4. Accurate temperature control
5. Adequate humidifying apparatus
6. Adequate humidity control

Air cleaning devices have been omitted in the 95 per cent equipment as this percentage can be obtained under ordinary conditions without air washers or filters. In an exceptionally clean locality, such high percentages can be obtained.

Class C—90 per cent equipment. A mechanical supply system consisting of the following:

1. An adequate air supply with 30 c.f.m. per occupant
2. Gravity exhaust
3. Efficient air distribution
4. Automatic temperature control
5. Adequate humidifying apparatus
6. Humidity control in the main duct only or from a typical room

Class D—85 per cent equipment. A mechanical system consisting of the following:

1. An accurate air supply with a maximum capacity of 30 c.f.m. per occupant
2. Gravity exhaust or exhaust openings
3. Good air distribution
4. Automatic temperature control

Class E—80 per cent equipment.

1. A positive air supply with gravity exhaust but without air-cleaning devices, humidifying apparatus, temperature or humidity control
2. Direct-indirect systems with either mechanical or gravity exhaust
3. Open window or other so-called natural systems of ventilation

Note: Synthetic Air Chart—The final form of the Chart and the text will be revised and comfort based upon the equivalent temperature curve rather than on the wet bulb.

Some Practical Standards of Ventilation

Turning back to the ten factors previously given under the theoretical standards of ventilation, it will be seen that these might be grouped as follows: Air supply and air distribution under one head which might be termed, effective air supply.

Effective air supply would then represent the amount of air actually brought into proper relationship with the occupants of the ventilated space. It is said that an adult could exist indefinitely on the quantity of oxygen contained in 5 cu. ft. of air supplied per hour if the lungs could extract all of this oxygen. The practical percentage of extraction is placed at 25 per cent so that 20 cu. ft. of air per hour will sustain life. An adult at rest expires about 0.6 cu. ft. of CO_2 per hour so that with 20 cu. ft. supplied per hour the CO_2 content would be 300 parts in 10,000. This equals the 300 parts fixed by the Synthetic Air Chart for zero ventilation.

If a deduction of 0.3 per cent is allowed for each part of CO_2 in 10,000 and then is divided by the distribution percentage, we would get the deduction percentage for what might be termed the effective air supply column, covering both of these factors. For example, if the CO_2 test showed 6 parts above the outside air standard of 4 parts in 10,000 we would have according to the Synthetic Air Chart a deduction for CO_2 of $6 \times 0.3 = 1.8$ per cent. If for this condition the test showed a distribution of 50 per cent, we would have $1.8/50$ per cent = 3.6 per cent as the combined deduction for these two factors in the effective air supply column. This would be the same as would be got in the CO_2 column alone with the alhalr sup-

ply and perfect distribution which is as it should be, as the usefulness of the air supply would be the same in either case. This would simplify the chart and put the measure of these two factors on a more scientific basis.

It has already been seen that air temperature, relative humidity and air motion naturally combine under what has been termed effective temperature.

Remaining then are the following practical factors of ventilation, stated in the order of their importance.

1. Effective air supply
2. Effective air temperature
3. Air cleanliness
4. Air sanitation
5. Odors
6. Other injurious substances
7. Monotony

Taking these as a basis and including the other suggestions made, the Synthetic Air Chart might be put into the following simplified and perhaps more usable form.

A further suggestion is, that since the percentages of deduction for the particular columns, or departments, representing effective temperature and effective air supply are so nearly the same as the corresponding deductions to be carried over into the final percentage of perfection column, that these be made exactly the same, for the sake of simplicity and ease of use. Another suggestion is that a function on effective temperature distribution be added similar to that suggested for the effective air supply.

In the normally crowded city place of assemblage the heat given off by the occupants together with that given off by the lighting and power equipments is usually more than the normal heat loss through the structure to the outside air, even in winter under cold climatic conditions. This means that in order to preserve an equilibrium of effective temperature the entering air must be cooler than the leaving air, so that the problem is usually one of cooling and ventilating rather than of heating and ventilating.

A typical case for winter might show about 300 B.t.u. of body heat plus 100 B.t.u. from light, etc., being given up to the building against 200 B.t.u. heat loss from the building, per person per hour. This would mean that 200 B.t.u. per person must be carried away by the air.

If the flow of air is upward, or from the side, so as to bring the incoming air into direct contact with the occupants, the temperature of the incoming air may not be more than 5 deg. below the temperature of the air leaving the occupant (for ceilings 10 ft. or less in height) otherwise the ventilation will be drafty and uncomfortable. This difference may be increased 1 deg. for each 2 ft. of added ceiling height provided the rising air does not come into direct contact with another tier of occupants. For 10 ft. and lower ceilings the quantity of air per person to dissipate this excess heat is $200/60 \times 0.02 \times 5 = 33$ cu. ft. per person per min. This may be reduced somewhat on the assumption that the component of heat from lights is usually introduced near the ceiling and may be allowed to heat the outgoing air to a greater difference.

In the practical work of engineers who design ventilating systems and of architects and owners who have to pass upon these systems, the one item involving standards which is the basis of all calculations and layouts, is the quantity of air to be

lution only, the following cubic feet per minute per person would be required for the ventilation percentages shown, if all other factors are 100 per cent perfect.

Method of Using Modified Synthetic Air Chart

The accompanying chart, Fig. 2, (on opposite page) represents the same conditions as shown on chart Fig. 1, corresponding to the following test data:

Dry bulb temperature.....	72 deg.
Wet bulb temperature.....	58 deg.
Air motion.....	20 ft. per min.
Physical state.....	Light work
Dust.....	10,000 particles per cu. ft.
Bacteria.....	10 colonies on a 2 min. plate
Odors.....	90 per cent free from
CO ₂	7 parts per 10,000
Other injurious substances.....	None
Distribution.....	81.4

The values are plotted on the modified chart as follows: Referring to the Comfort Chart shown between pages 134-135 of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS' GUIDE, 1923, it will be seen that a dry bulb temperature of 72 deg. with a wet bulb temperature of 58 deg. shows an effective temperature of 65.25 deg. It will be seen by referring to Fig. 10 on page 180 of the February 1924, JOURNAL AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS that the air motion of 20 ft. per min. would reduce this effective temperature about 0.75 deg. Assuming from previous data that the state of light work would reduce the comfort line $1\frac{1}{2}$ deg. it would bring this to 62 $\frac{1}{2}$ deg. instead of 64 deg. This will leave an effective temperature difference of 2 deg.

This corresponds to a deduction of 5 per cent for this particular column leaving a plus percentage of 95 per cent. This 5 per cent deduction in the effective temperature column corresponds to a deduction of 4.5 per cent in the final percentage of perfection column.

The 10,000 particles of dust corresponds to 96 per cent in the plus column for this particular department and a minus 1 per cent to be deducted in the final percentage of perfection column.

The 10 colonies of bacteria correspond to 98 per cent in the plus column for this department and a minus 1 per cent to be deducted in the final percentage of perfection column.

The 90 per cent freedom from odors correspond to 94 per cent in this particular column and a deduction of 1.5 per cent for the final percentage of perfection column.

The seven parts of CO₂ (3 parts above the normal CO₂ contents of the outside air) correspond to a plus percentage of 99 per cent for this particular department and a deduction of 0.9 per cent in the minus column.

This 0.9 per cent is extended over to the diagonal line corresponding to 80 per cent distribution and from the intersection of these two lines a perpendicular is dropped to the line showing the combined minus per cent for CO₂ and distribution, which is found to be 1.15.

Adding these several deductions we have the following:

For effective temperature difference	minus 4.5%
For dust,	minus 1%
For bacteria,	minus 1%
For odors,	minus 1.5%
For effective air supply,	minus 1.15%
Total	9.15%

This deducted from 100 per cent leaves 90.85 per cent which is plotted in the final percentage of perfection column.

In theatres, assembly rooms, auditoriums and other places of public amusement and assemblage there are usually several other factors to consider, such as the removal of excess heat, excess moisture, dust raised by the movement of the occupants and odors.

Percentage of Perfection	Cu. ft. of Air Per Minute Required Per Person at Rest	Cu. ft. of Air Per Minute Required Per Person at Hard Work
98%	15.0	30
96%	7.5	15.0
94%	5.0	10.0
92%	3.75	7.5
90%	3.0	6.0

For 85 deg. outside air and 70 per cent relative humidity in the summer the effective temperature percentage of perfection would drop to 60 per cent without any change in the air from outside conditions. Assuming that the distribution was 85 per cent perfect the total deduction would be 40/80 per cent = 50 per cent leaving a plus percentage of 50 per cent for this column. This corresponds to 45 per cent reduction in the final column, leaving 55 per cent perfection of ventilation for any amount of air supply, provided all other departments are 100 per cent.

Assuming that the air supply is 30 cu. ft. per person and that the body heat and heat from equipment will raise this 5 deg.; also that the vapor added per person is 10 grains per minute or 0.33 grains per cu. ft. of air handled, the effective temperature difference will be raised about $2\frac{1}{2}$ deg., so that the percentage of ventilation, would drop about 6 per cent more, leaving 49 per cent ventilation. It will be seen therefore, that we may not hope to get better than 40 per cent to 60 per cent ventilation in warm summer weather without some method of air cooling.

Air motion will assist but unless increased beyond the usual 10 to 20 ft. per min. ordinarily obtained from the movement of the air through the room it will not improve the percentage of ventilation more than 2 to 3 per cent.

By the use of refrigerating and dehumidifying apparatus this effective temperature department can be maintained at any desired percentage of perfection. The use of a good air washer should reduce the temperature about 70 per cent of the difference between the wet and dry bulb temperature. This for the above case would reduce the effective temperature difference about 2 deg. corresponding to an increase of 5 per cent in the final percentage of the ventilation.

It will be understood that the above is rather an extreme case of temperature and humidity and that the final percentage will be improved by the air washer in a greater proportion if the relative humidity of the outside air is lower.

For a condition of 80 deg. dry bulb and 50 per cent relative humidity, the percentage for the entering air would be 79 per cent, the percentage leaving the occupants would be 67.5 per cent and the air washer would improve this to 76 per cent.

Assuming that 90 per cent ventilation is desired for places of assemblage, that distribution will be 75 per cent, dust 96 per cent, bacteria 98 per cent and odors 85 per cent, there will be a deduction of 1 per cent for dust plus 1 per cent for bacteria plus 1.5 per cent for odors making a total of 3.5 per cent and leaving a deduction of 6.5 per cent for effective temperature plus effective air supply.

Assuming that the effective temperature can be controlled in winter to within 1 deg. above or below the comfort line there would be a deduction of $2\frac{1}{2}$ per cent for this, leaving a deduction of 4 per cent for effective air supply. With 75 per cent distribution this would leave 3 per cent deduction for CO₂ corresponding to an air supply of 10 cu. ft. per min. per person.

For a summer condition of 80 deg. dry bulb and 65 per cent relative humidity without air cooling the effective temperature percentage would drop to 74 per cent for the entering air and 52 per cent for the air leaving the occupants. This would be raised to 66 per cent with the air washer.

For 30 cu. ft. per person per minute these percentages would change to 74 per cent for the entering air, 66 per cent for the air leaving the occupants and 76 per cent with air washers.

For 40 cu. ft. per person per min. these values would change to 74 per cent, 67 and 80 per cent, and for 50 cu. ft. per person to 74 per cent, 68 and 81 per cent.

It will be seen from this that 10 cu. ft. of air taken in from the outside, per person per minute, is sufficient for winter conditions but inadequate for summer climates, where heat and humidity are the determining factors, unless refrigeration is used. It should also be noted that increasing the air supply from 10 to 50 cu. ft. per person per min. gives little improvement unless some form of artificial cooling is used.

It would seem then that about 75 per cent ventilation can be obtained under reasonably severe summer conditions with an air supply of 30 cu. ft. per person, using an air washer, and that beyond this point there is little to be gained by increasing the supply. It should be noted that the above is based on upward ventilation and that the cooling effect of from 5 to 10 deg. with air washers and of perhaps twice this amount with refrigeration will produce uncomfortable drafts on the occupants at times. For this reason and for the further reasons that it is more sanitary and more easily controlled, the downward system of ventilation is perhaps more efficacious in large and intensely used places of assemblage. On account of transporting all of the heat from lights downward and of forcing the body-heated air back over the occupants it is usually necessary to do much more cooling of the air than can be done with the air washer without refrigeration.

On the other hand the air is brought in high enough to permit of its being diffused and brought to the proper condition before coming into contact with the occupants. It can be seen, therefore that the air supply per person per minute for assemblies could be 10 cu. ft. in winter, 30 cu. ft. in summer with air washers and anywhere between these two figures for the entire year with refrigeration. Also that nothing better than about 75 per cent ventilation can be secured in hot sultry summer weather without artificial cooling but that with such cooling, especially if the air supply is taken from overhead and exhausted from below, most any percentage of perfection can be maintained.

The foregoing does not take into consideration the matter of recirculation, but it can readily be seen that there is little to be gained by recirculation unless an appreciable amount of CO₂ and attendant impurities which are put into the air, can be taken out during recirculation. The handling of the larger quantity of air may be of value either to produce air motion or for use as a better cooling medium with less temperature difference between incoming and outgoing air. Recirculation may also be used as a purely economic feature during the warming up of the building or during periods when the space is only partly occupied and the mechanical arrangements are inadequate for properly varying the quantity of air handled to suit.

A good arrangement is to provide apparatus for handling 30 cu. ft. of air per person per minute with provisions for recirculating any amount up to as much as $\frac{2}{3}$ of this. The percentage of air recirculated may be varied to suit the seasonal change so as to conserve heat in winter and refrigeration in summer.

Where effective temperature is controlled, according to the usual method, from the dry bulb temperature in the room there may be a wide variation in this effective temperature due to the varying amounts of moisture in the air, unless humidifying apparatus with accurate humidity control is employed.

Between the condition of absolute dry air at 70 per cent and absolutely saturated

air at 70 deg. there is a difference of 10 deg. in effective temperature which means an average difference of 25 per cent in the quality of ventilation. This may be taken to mean about 10 per cent on each side of the neutral point for ordinary ventilating conditions so that the air washer and humidity should improve the ordinary ventilating plant another 10 per cent on this count. Good filters will of course serve the same purpose for cleaning the air of suspended matter, and may improve the ventilation about 10 per cent.

In schools where the requirements are not so severe and it is not necessary to provide for summer conditions as much as 90 per cent of the air is being very successfully recirculated. The use of ozonation for eliminating odors and for otherwise refreshing the air is advisable wherever recirculation is regularly and intensively employed.

It should be noted in connection with our present means of measuring and comparing qualities of ventilation that we do not take into account any of the functions of the relative humidity of the air except that bearing upon effective temperature. This means that air of any temperature and relative humidity, within proper physical range, *i. e.*, below 64 deg. wet bulb, may be made to meet the comfort line by either heating or cooling without addition or deduction of moisture. Absolutely dry air may be heated or cooled to 78 deg. and be 100 per cent perfect as far as effective temperature is concerned and still be far from perfect as far as effects on the membranes of nose, throat and lungs are concerned. Such dry air is also very conducive to the increase of dustiness in the atmosphere of a room from the stand-points of dryness and electro-static agitation. The air washer and humidifier correct these difficulties and there should be some definition of limits for the relative humidity in our measure of ventilation.

It is not unusual to find from 1 to 2 million particles of dust per cubic foot in the outside air surrounding city buildings and unless this is eliminated it will give dust counts in rooms equivalent to a deduction of from 5 to 20 per cent in the perfection of ventilation.

A good air washer should eliminate 80 to 90 per cent of the dust entering the intake and perhaps reduce the dust penalty in the rooms to less than one half of the above figure. It will be seen, therefore, that air washing and humidification may improve the quality of ventilation about 10 per cent in the effective temperature department, plus another 10 per cent in the dust department, plus other improvements in the quality of ventilation by maintaining proper humidity and removing other injurious substances and odors.

When intensive recirculation is employed the use of milk of lime in the washer water may be employed to assist in the removal of CO_2 from the recirculated air.

In connection with the ventilation of schools the condition in the class rooms for average winter conditions is that the body heat given up to the room from the occupants is less than the heat loss from the building to the outside air so that the incoming air may be maintained at a higher temperature than that of the air surrounding the occupants. This means that the effective temperature may be controlled within 1 deg. above or below the comfort line.

Assuming that 95 per cent ventilation is required for class rooms, that the distribution is 85 per cent and there is no deduction for dust, bacteria or odors, we could have a deduction of 85 per cent of 2.5 per cent which is equivalent to about 2 per cent for CO_2 . This would mean 15 cu. ft. of air per pupil per minute.

The usual requirements of state laws is 30 cu. ft. per pupil which would mean a deduction of 1 per cent for CO_2 leaving a 4 per cent deduction for effective tempera-

ture, dust, bacteria and odors. Allowing a deduction of $2\frac{1}{2}$ per cent for effective temperature there would be a possible deduction of $1\frac{1}{2}$ per cent for these other items. It will depend therefore upon the quality that can be maintained for these other items as to the actual quantity of air required between 15 and 30 cu. ft. per minute per pupil.

Where intensive recirculation is employed the extent of the recirculation will depend on the quantity of CO_2 , odors and other objectionable factors which can be removed from the recirculated air and also upon the ability to keep the relative humidity from rising to an undesirable point on account of the vapor given up to the air by occupants of the building. There is no question but that with proper apparatus and operation recirculation in class rooms can be successfully carried to from 75 to 85 per cent where 30 cu. ft. of air per minute is handled per pupil.

In the ventilation of school auditoriums where the occupancy of the rooms is of relatively short duration of from 1 to 2 hours the initial air in the rooms may be relied upon to reduce the intensity of ventilation required so that from 15 to 20 cu. ft. of air per person per minute is usually sufficient.

School toilets should be separately ventilated with an air change of from 2 to 5 min. employing mechanical supply and exhaust with the exhaust 20 per cent in excess of the supply in order to prevent objectionable odors from diffusing into other parts of the building.

Kitchen and lunch rooms should be ventilated with about a 5-minute air change for lunch rooms and a 1- to 3-minute air change for kitchens with at least a part of the exhaust taken from the lunch rooms through the kitchen so as to keep all of the air traveling towards the kitchen, thus preventing the kitchen odors from diffusing into the lunch rooms.

In the ventilation of hospitals the wards may be treated much the same as the class rooms of a school with the exception that it is inadvisable to use recirculation on account of the danger of contagion.

Toilets should be ventilated separately the same as for schools.

Dining rooms, kitchen and diet kitchens should have double mechanical ventilation giving from 3- to 5-min. air changes for dining rooms, from 1- to 3-min. changes for kitchen with the exhaust about 20 per cent in excess of the air supply with at least a part of the dining room exhaust passing out through the kitchen so as to keep all of the air travel toward the kitchen and prevent diffusion of odors throughout the building.

Dining rooms, kitchen and toilet exhaust should be discharged above the roof of the building.

Exhaust from kitchen range hoods should be discharged by a separate exhaust fan through a fire proof metal duct extending above the roof of the building provided with automatic fire damper and steam jet fire extinguisher for use in case of emergency, as the accumulation of the grease vapors in this flue frequently causes fire.

Private rooms if ventilated should be provided with double mechanical supply and exhaust ventilation as either a supply without exhaust or exhaust without supply will tend to force or draw the contaminated air from one room to another.

Hotels, kitchens, restaurants and dining rooms should be ventilated the same as for hospitals.

It would be interesting to analyze the ventilation for other classes of buildings if time and space were available but in lieu of this the following table is submitted

in an endeavor to outline the kinds of spaces usually requiring ventilation in various classes of buildings together with the quantity of air ordinarily used for securing good ventilation results.

TABLE 1. CUBIC FEET OF NEW AIR TO BE SUPPLIED PER PERSON PER MINUTE

	Without humidification or recirculation	With humidification but without recirculation	With humidification and recirculation	Number of air changes per hour
SCHOOLS				
Class Rooms	30	20	5 to 10	
Assembly Rooms	15 to 20	10 to 15	5 to 10	
Gymnasiums	30	25	15 to 20	
Toilets				10 to 20
Locker Rooms				5 to 10
Kitchens				20 to 60
Lunch Rooms				10 to 20
THEATERS				
Seating Space	30 to 50	20 to 30	10 to 15	
HOSPITALS				
Wards	30 to 40	20 to 30		
Kitchens				20 to 60
Dining Rooms				10 to 20
Toilets				10 to 20
HOTELS				
Dining Rooms				10 to 15
Kitchens				20 to 60
Ball Rooms				5 to 10
Work Space				5 to 10
Assembly Rooms	20 to 30	15 to 20	10 to 15	

APPENDIX

Special attention is called to the fact that all that has been said regarding the quality of ventilation, as effected by effective temperature for Summer conditions, is based on our present standard of a 64 deg. comfort line, applicable to all seasons of the year.

There is no question that the seasonal differences in the out of doors atmospheric conditions, the difference in the weight of one's clothing between Winter and Summer and the corresponding physiological reactions between being out of doors and indoors should have considerable bearing upon the comfort line.

In the measuring of the quality of ventilation we should no doubt, therefore, use different effective temperatures for comfort for the different seasons of the year and it may be that the comfort line should be made a function of the outside effective temperature.

No. 709

SOME COMMENTS ON PRESENT DAY HEATING AND VENTILATING PRACTICE

By W. S. TIMMIS,¹ NEW YORK, N. Y.

MEMBER

THE influence of heating and ventilation in our everyday activities will be discussed briefly, and an attempt will be made to outline the results of recent investigations in heating and ventilating research. General data will be given on the physical requirements for heating and ventilation as determined by the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS; reference will be made to the investigations of the New York State Commission on Ventilation, also to the analysis of its report made by a Special Committee of the Society; and consideration will be given to the various kinds of heating systems now in common use. In the discussion of these subjects the factors that will be included cover properties of air, effect of humidity and air movement, maximum temperatures, psychology of the open window, heating requirements, infiltration and radiation.

The Research Laboratory of the A.S.H.&V.E. is working in conjunction with the U. S. Bureau of Mines in Pittsburgh and is fully equipped with a competent staff of physicists and scientists who are working out the underlying problems of this general subject and are giving attention to the development of fundamental data for the use of engineers, architects, contractors and the public in general who are interested in the science and application of heating and ventilation.

For many years the thermometer reading separate degrees on the fahrenheit scale has been regarded as the guide for temperature conditions, and as registering a comfortable degree of temperature irrespective of humidity, air movement, persons at rest or working, engaged in either sedentary occupation, or in active work. The thermometer which is used is, in reality, only partially successful in giving an indication of the proper atmospheric condition. From a large number of tests which have been made, it has been ascertained that a comfortable air condition is one involving higher temperatures when the air is dry than when the air is moist, and higher temperatures when the air is moving than when it is stationary. The effect of humidity on comfort is readily understood by referring to the Comfort Chart Fig. 1. It will be seen that the Comfort Zone is quite broad with humans at rest, and that the average person can be quite comfortable within that zone. From this chart it will be seen that people are quite as comfortable with a wet bulb tem-

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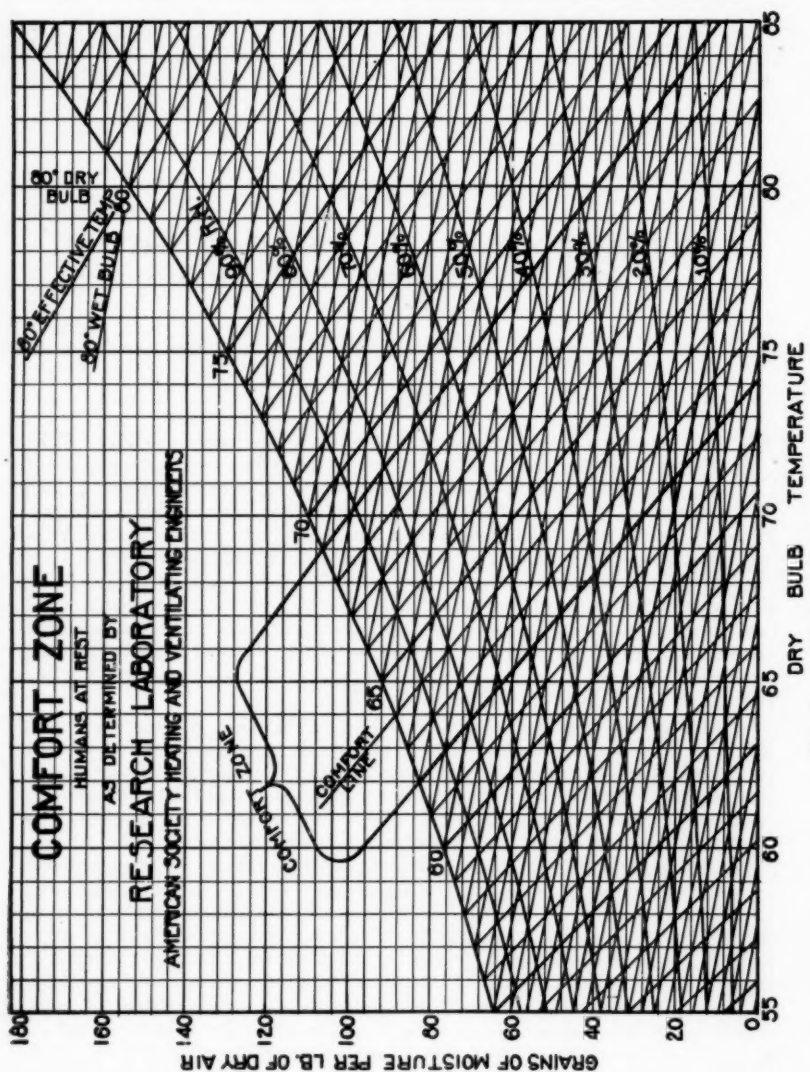


FIG. 1. COMFORT ZONE FOR HUMANS AT REST

perature of 64 deg. fahr., and the air saturated, as with a 70 deg. temperature on the dry bulb, with 57 deg. temperature on the wet bulb, which would give a percentage of moisture of about 45 per cent, and the same degree of comfort can be experienced with 76 deg. dry bulb temperature with 10 per cent moisture.

The wet bulb thermometer, known as a psychrometer, consists of two thermometers, one of which is covered with a piece of silk moistened with clean water. By revolving this psychrometer, air motion can be produced on the wet bulb thermometer and the water in evaporating will abstract a certain amount of heat from the wet bulb. A lower temperature will be shown or a difference in temperature between the dry bulb and the wet bulb which is a measure of the humidity. The percentage of humidity can be found by referring to the Comfort Chart. The other chart (Fig. 2), will give an idea of the effect of air movement and shows conclusively that, for the same comfort, higher temperatures are necessary when air is in motion.

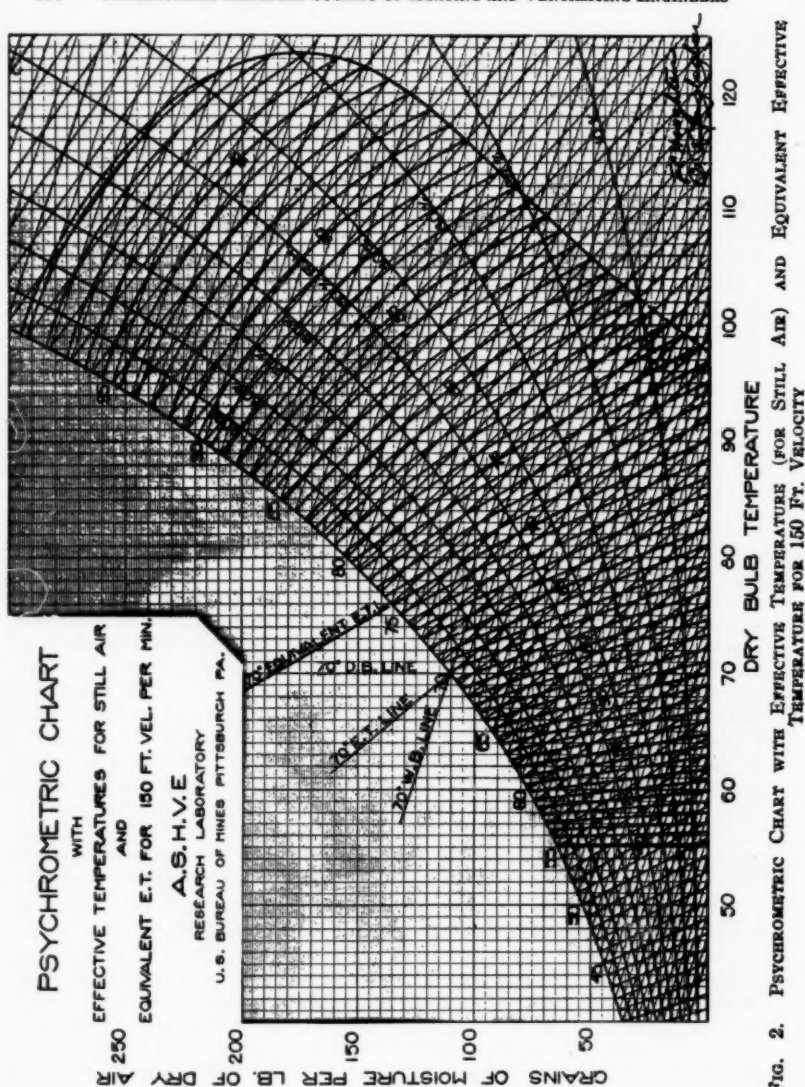
Perhaps the best definition of ventilation is the one following, given by the AMERICAN SOCIETY HEATING AND VENTILATING ENGINEERS:

Ventilation perfection is attained when the atmospheric condition in every part of a room with proper air motion occupied by human beings is continually maintained with normal amount of oxygen and free from dust, bacteria, odors and poisons, and the temperature and humidity quality shown within the comfort zones, as determined by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS Research Laboratory, and that the apparatus or method which most nearly causes atmospheric air to meet this ventilation perfection or ideal is the most satisfactory.

The New York State Ventilation Commission has made a very exhaustive report covering its work of several years on the requirements for ideal atmospheric conditions. This report is, of course, a voluminous document and cannot be considered at great length here. It is shown, however, that people work much less effectively in a hot humid room than in a relatively cool and dry atmosphere; that they accomplish 28 per cent less physical work in a temperature of 86 deg. fahr. with 80 per cent relative humidity than in a room of 68 deg. fahr. with 50 per cent relative humidity. Their tests establish the fact beyond question that high temperatures, particularly when combined with high humidity reduce the capacity for physical work. On the other hand, the interesting fact is disclosed that purely mental work is only slightly affected by a rise in temperature from 68 deg. to 75 deg. The Commission found that a marked influence was exerted by stale air upon the appetite for food, and that, under fresh air conditions, the amount of additional food consumed ran from $4\frac{1}{2}$ per cent to $13\frac{9}{10}$ per cent in the various tests made. The amount of air required varies from 20 to 30 cu. ft. per minute per person.

It is highly essential to have air movement, as stagnant air is depressing and variable temperatures, as in the open air, seem to be also desirable, that is, the movement of different temperatures past the body seem to give the best feeling of relief and comfort. Such movement of air is provided automatically in most homes during a high wind, for it is found that badly fitting windows provide change of air in a home from two to four times per hour, in other words, there is a complete air change every 15 to 30 min. Windows that fit properly reduce the amount of air leakage 50 per cent, and such windows properly weatherstripped in metal make a further 50 per cent reduction.

The amount of ventilation in various classes of buildings varies, as, for instance, in hotel laundries it is necessary to have a change of air about every 3 min., and in hotel kitchens every 4 min. for supply, and essential to remove the air about every 3 min., whereas in boiler rooms it is necessary to make an air change about every 3 min., with an exhaust about every 4 min.



Any room or building used for the habitation or congregation of human beings should be provided with a plentiful supply of fresh air. Strictly speaking, good ventilation is merely a relative term, and the standards as ordinarily accepted are a compromise that will answer the purpose of keeping the air in a building in a fairly fresh condition. The requirements of ventilation are; *First*, to maintain cer-

tain standards of purity of the air within the room or building; *Second*, to remove and prevent odors; *Third*, to remove the body heat of the occupants and the heat from such other sources as illumination and power; and *Fourth*, to prevent excessive rise in humidity which usually accompanies the rise in temperature from bodily heat.

Many of the existing standards of ventilation have been founded on the belief that carbon dioxide was the dangerous element in expired air. The requirements of ventilation in air purity are more or less arbitrary, and no rational standard has ever been fixed. Later investigations would indicate that carbon dioxide is harmless, and interesting only as indicating how much respiration the air has undergone. In this way, it serves as a test of the contamination of the air by organic impurities from the lungs and bodies of the occupants. These organic poisons are little understood, although they undoubtedly constitute the real danger in impure air. The standard of purity which has usually been considered satisfactory is from six to eight parts of carbon dioxide in 10,000 parts of air, but it is certain that ten times this amount would not be injurious if provision were made for the removal of organic impurities. In all probability the best indication of good ventilation insofar as purity is concerned is freedom from objectionable odors.

It is estimated that the average adult, at rest or doing light work, will breathe approximately 0.25 cu. ft. of air and exhale 0.01 cu. ft. of CO_2 per minute (0.6 cu. ft. per hour), and that only about 5 per cent or less of the oxygen is taken out of a breath of air. The air of poorly ventilated rooms will show a slight diminution in the oxygen, accompanied by a corresponding increase in carbon dioxide, organic pollution, and moisture. The poisons in the air due to the presence of too many persons relative to the supply induce a lowering of the vital processes, and a loss of muscular strength.

Ordinary outdoor air will contain on an average about four parts of CO_2 in 10,000 parts of air and good ventilation is ordinarily considered to exist in a room where the air contains not more than from six to eight parts of CO_2 in the same amount of air. That is, if a great amount of CO_2 exists in the air, it is considered as having been inhaled too much and unfit for further respiration. The following table gives the amount of air required per hour by the average person, exhaling 0.6 cu. ft. of CO_2 per hour, if it is desired to maintain the corresponding number of parts of CO_2 in the air with outdoor air containing four parts of CO_2 per 10,000 parts of air.

Parts of CO_2 in 10,000 parts of air:

Increase above Outdoor Air	Total	Cu. Ft. Air per Hr. per Person
1	5	6000
2	6	3000
3	7	2000
4	8	1500
5	9	1200
6	10	1000

It is ordinarily the custom to allow for average conditions 1800 cu. ft. of air per hr. per person, and this is the factor commonly used for school ventilation. But there are many cases in which the amount of air allowed is varied to suit the circumstances, a few of which are given in the accompanying table.

In rooms where the glass and wall exposure is considerable, ventilation for the removal of body heat need not be considered, except where the building is artificially cooled. In crowded audience halls, however, and even in school rooms it is the determining factor. Each adult occupant gives off an average of 400 B.t.u.

per hour, of which approximately 150 B.t.u. may be assumed to be latent heat of evaporation, while not more than 250 B.t.u. will be sensible heat given off by the breath, and by convection to the surrounding air. On this basis if each occupant is supplied with 30 cu. ft. of air per min. or 1800 cu. ft. per hr. there will be a rise of approximately 8 deg. above the temperature at which the air is introduced into the

AIR ALLOWED PER PERSON CU. FT. PER HR.

Hospitals (ordinary).....	2100 to 2400	Theatres.....	1200 to 1800
Hospitals (epidemic).....	4800	Meeting halls.....	1200
Work Shops.....	1500	Schools (per child).....	1800
Prisons.....	1800	Schools (per adult).....	2400

room, so that in order to maintain about 70 deg. in the room, the air would have to be reduced to 62 deg. There is evidently a limit to the difference of temperature allowable between incoming air and room temperature, which depends largely on the size and arrangement of inlet openings as effecting the production of cold drafts. The practical limit is found, in standard methods of ventilation, to be between 25 and 50 cu. ft. per min. per adult occupant. This, as may be noted, also gives a very satisfactory standard of purity.

While 1800 cu. ft. of air per hr. or 30 cu. ft. per min. (expressed as 30 A.P.M.), when used as a standard for overhead ventilation, is, in the average case, amply sufficient to take care of the heat and moisture from the body, when the air is supplied through many small openings distributed about the room, a smaller quantity of air may often be supplied. Several different systems of this character have been used, such as introducing the air under the seats in a theatre, or through a small opening at each desk in a school. By this means a more uniform distribution of the air is obtained than is possible with the over-head system, with greater assurance that each occupant of the room will receive the desired amount of fresh air.

Each cu. ft. of gas requires $8\frac{1}{2}$ cu. ft. of air.

Each lb. of oil requires 150 cu. ft. of air.

One cu. ft. of gas gives off approximately 600 B.t.u. per hr.

The effect of hearty diet containing hydrocarbons is to make it possible to live comfortably under much lower temperatures. One of the causes of inability to eat hearty food lies in the fact that the average home or place of assembly is so heated that the heat content of such food cannot be disposed of as the heat must be displaced from the body into a low enough temperature, and this explains why people living in northern climates without such a temperature differential as is generally maintained in homes can partake of foods rich in fats and maintain their health at the same time.

The effect of clothing is another important factor in the necessity for higher temperatures during the winter season, as the amount of clothing is reduced, the temperature requirements for comfort are necessarily higher, and there has been a gradual ascension of indoor temperature until now it is quite common for indoor temperatures in places of assembly, homes, and hotels to rise to a point of between 70 deg. and 80 deg. even when the external temperature is zero. It would be much healthier if there could be maintained a lesser differential in the winter time between the outdoor and the indoor temperature, and an adjustment made in modifying clothing and the diet.

Pure water is the chemical combination of H_2O formed by the union of 2 volumes of hydrogen with 1 vol. of oxygen gas. It expands heated to 39.2 deg. fahr., which is the maximum density, when raised to any higher temperature, but contracts heated from 32 to 39.2 deg. fahr., and boils at 212 deg. fahr. The specific heat of

water or the number of B.t.u. required to raise the temperature of 1 lb. of water 1 deg. varies slightly with the different temperatures.

Pure dry air is a mechanical mixture of oxygen and nitrogen, that is, the oxygen and nitrogen can be separated from each other by purely physical means. It also contains other constituents such as carbon dioxide, ozone, water vapor, dust, and bacteria. The specific density, or weight per cubic foot of dry air, decreases with the temperature, and the specific volume, or volume per pound, increases with the temperature. The specific heat of air at constant pressure, or the B.t.u. to raise 1 lb. 1 deg. varies from 0.2375 to 0.2430, and the value of 0.24 is recommended for general calculations.

Humidity is the water vapor mixed with the air. Saturated air is said to be saturated when it has mixed with it the maximum possible amount of vapor. The actual humidity of the air is taken as the number of grains (1 lb. = 7000 grains) or pounds of water vapor contained by 1 cu. ft. of a mixture of air and vapor, and the relative humidity is the percentage of saturation. At each different temperature, air will contain at the saturation point, varying amounts of moisture.

The dew point temperature corresponds to saturation, or 100 per cent relative humidity, and whenever the temperature of saturated air is lowered precipitation takes place to a point of saturation of the lowered temperature.

For many years attempts have been made to install what are known as closed systems of ventilation, that means to have the buildings entirely enclosed and permitting only such openings in the building as are necessary for ingress and egress, but there is a psychology to be satisfied, especially in educational institutions, which shows itself in the desire for the open window, and this is particularly true when the temperatures become excessive, and regulation of temperature and humidity are important. It may be called a rebellion against being choked by too much heat and a desire for the refreshing effect of mixed temperature and air movement, such as one gets from the outside atmospheric condition. Open window ventilation, however, is not altogether satisfactory, and, if indulged in without great care, will be productive of evil results.

Heating systems in common, outside of the family stove, consist of warm-air furnaces, steam heating and hot-water plants, and forced air circulation. Many of these systems may be sub-divided, as for instance, in steam heating plants there are what is known as gravity heating, both single and two pipe systems, vapor-heating and vacuum systems. Hot water systems are usually composed of a two pipe system, that is, a supply and return.

As to the desirability of the various systems for any particular case, many factors will have to be taken into consideration, such as the element of first cost, economy of fuel, and cost of operation.

The heating requirements for buildings shall be such as to take care of infiltration, that is, air movement into a building and heat leaving the building by means of air movement; radiation, which is the radiating heat given by the building to the external temperature; and the heating of the necessary amount of air for ventilation where such is provided.

The amount of infiltration will depend upon the nature and tightness of the building construction, the number of feet of window and door cracks, and the presence of open fireplaces and vent registers, and the air admitted by opening and closing of doors, and other openings. This loss can be materially reduced by the use of storm doors and windows, metal weatherstrips on doors and windows and, in frame construction, by a closely fitting tongue and groove siding covered with

heavy building paper. It is not generally known that there is a considerable loss by leakage through brick walls, and that there is a greater loss through hollow tile.

The effect of high wind velocity in the heating of buildings is very marked, and it has been estimated that it is as difficult to heat an exposed room at 20 deg. above zero with a 35 mile wind blowing, as it would be to heat the same room at 20 deg. below zero with no wind blowing. Fortunately, strong winds very seldom prevail at temperatures below 15 deg. above zero.

The computation of the amount of radiation required for a building is comparatively simple as each of the elements of building construction have been found to radiate varying amounts of heat, and the problem for the engineer is to take the area of the wall, windows, doors, in fact all external exposure, including roof, and multiply such area by the various factors which have been determined, then add to this amount, the heating requirements for the ventilation and infiltration, thus obtaining the total amount of heat in thermal units per hour which will have to be supplied by the heating system. Generally speaking, it may be said that this total number of thermal units, when divided by 250 will give the number of square feet of steam heating surface required, but modifications of this figure of 250 will have to be made depending upon the type of radiation selected, whether single, 1-, 2- or 3-column, and the kind and length of the radiators, all of which have a bearing on the number of thermal units given off per square foot of radiation.

The effect of metallic paint, such as aluminum or gold bronze, upon radiation has been the subject of investigation, and it has been discovered that for a given radiator the number of thermal units given off per sq. ft. of bare cast iron is 240, and the same radiator painted with aluminum bronze would only give off 200 thermal units, and if then painted with white enamel would give off 242 units, and if painted again with bronze aluminum would bring it down to 200, and painted again in gold bronze would raise it to 205.

The catalog rating of furnaces, boilers and radiation have to be scrutinized very carefully, as in the hands of the uninitiated, these ratings will be misleading. Most of the rating for boilers should be greatly reduced to take care of operating conditions for the radiation required, and for the steam piping which is not included in the catalog ratings, and also to take care of the operating conditions of the boiler which is not always kept clean. Generally speaking, it will be necessary to secure a boiler in capacity 100 per cent greater than the number of square feet of radiation. It is much better to buy boilers on the B.t.u. basis than the sq. ft. of radiation basis. As a matter of fact, ultimately, radiation will be sold not upon a square foot basis, but on the basis of the number of units of heat given off by the radiation.

It is highly essential that greater control of temperatures be obtained in connection with heating plants. This control may be secured at the boiler to a limited extent in the case of steam heating and hot water systems by controlling the draft and consequently the pressure or temperature, but much remains to be done in the way of controlling temperature at the radiators, although we have some excellent systems of thermostatic control operating to shut off the steam when a predetermined temperature has been attained in the room.

While much has been accomplished in the improvement of heating and ventilating apparatus and practice during the past few years, there is room for much research on vital problems and the program now before the Laboratory of the Society presents possibilities for tremendous discoveries in this field of investigation.

IN MEMORIAM

	JOINED THE SOCIETY	DIED
RHEINHOLD F. LINDEMAN	Nov. 1916	Dec. 1923
WILLIAM J. BALDWIN	1915	May 1924
C. H. BLACKWELL	Sept. 1921	Apr. 1924
R. BENKENDORF	Sept. 1921	Apr. 1924
FRANK K. CHEW	1895	Mar. 1924
R. K. COFFIN	June 1921	Jan. 1924
ALLEN E. CUSTER	May 1921	Oct. 1924
J. J. FINAN, JR.	Dec. 1920	Oct. 1924
CHARLES GLENNON	Aug. 1917	July 1924
ROBERT D. HOPKINS	Dec. 1915	June 1924
HENRY LIPKEMAN	July 1915	June 1924
E. R. PIERCE	June 1919	Nov. 1924
R. L. REDPATH	July 1924	July 1924
GEORGE WEATHERWAX	June 1923	Mar. 1924

William J. Baldwin

The death of William J. Baldwin, Brooklyn, N. Y., brought to a sudden close the life of a man well known in the engineering field and an honorary member of the Society. His death came very unexpectedly on May 7 and was learned with sincere regret by all who were privileged to know him.

A native of Ireland, in which country he was born June 14, 1844, Mr. Baldwin spent the major part of his life in this country where he received his engineering education and training. He met with unusual success in his profession, and at the end of more than 79 years he had a very notable reputation as a result of his various successful undertakings.

As a naval constructor he did a great deal for the United States Government during the Civil War, and later the Brazilian Government employed him in the construction of naval vessels for that country. He was also one of the pioneer constructors in the engineering industry during the early period of New York's high building development, and one of the earliest evidences of his work in this particular endeavor is The Tribune Building in Park Row. Also as a consulting and designing engineer, he constructed the War College in Washington, D. C., and the Immigration Station in New York City, along with other Federal buildings. He also supervised the laying of the first telephone cable from New York to Brooklyn for the New York Telephone Co.

Mr. Baldwin was elected an honorary member of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS at the Annual Meeting, 1915.

He was also a life member of the *American Institute of Architects*, and had been a member of the *American Society of Mechanical Engineers* since 1883, a member of the *American Society of Civil Engineers* since 1884, and was a Telephone Pioneer of America, and a member of the Committee of the A.S.M.E. in 1886 when the Briggs Standard for Pipes and Pipe Threads was recommended and adopted as the United States Standard, since known as the American Standard.

Mr. Baldwin was also a member of an international commission which attempted to produce an acceptable International Standard for Pipe Threads from 1906 to 1914 when the World War interrupted the work temporarily.

For some time Mr. Baldwin served as Consulting Professor of Thermal Engineering at the Polytechnic Institute of Brooklyn, Brooklyn, N. Y., and in addition he was connected with *The Engineering Record* as its associate editor.

Not only was he editor but also author. He wrote and published a number of books on heating and ventilation which are widely used. *Baldwin on Heating, Thermus Papers, Hot Water Heating and Fitting, Data for Heating and Ventilation, An Outline of Ventilation and Warming, The Ventilation of the Schoolroom* are among his best known writings in the heating and ventilation fields.

He was also a contributor to the JOURNAL and several of his papers were reprinted for distribution in pamphlet form. "An Improvement in Ventilating Apparatus" and "Factory Heating from Fuel-Saving Angle" which he wrote in collaboration with R. C. Taggart are well-known among the latter.

Mr. Baldwin is survived by his wife and six children; three sons, William J., Jr., John R., and Thomas B. Baldwin, and three daughters, Mrs. Anna Harrington, Mrs. Rose Crosby, and Mrs. Josephine Mayer.

Mr. Baldwin's death was mourned not only by his immediate friends and relatives, but by a host of persons who have had contact with and associated with him in

the engineering world. All feel that the engineering profession has suffered an irreparable loss. The Society feels deeply the loss of its distinguished honorary member.

Frank K. Chew

The Society suffered the loss of one of its oldest and most active members in the death of Frank K. Chew, March 8, 1924. He was a pioneer in the heating and ventilating industry in which he spent more than 40 years of an active business life.

Earnestly devoted to the industry which he had made his business, he had watched with keen interest the steady advances in the heating, ventilating, and plumbing fields and had eagerly furthered any new steps and improvements. He gave the energy of his vigorous personality to the best interests of the trades he followed, and has left the imprint of his vitality and force upon the records of their progress. A legion of friends in all parts of the country will recall his unselfish and tireless service to his calling.

Mr. Chew was born in Salem, N. J., February 2, 1858. He first started in the heating and ventilating industry as a mechanic in a heating, plumbing and sheet metal shop in New Jersey. From this he went to the Abram Cox Stove Co. where he served for ten years as a salesman.

He then made a radical change in his line of work, going to Philadelphia to start in the field of trade journalism. He opened his editorial career with the *Metal Worker*, beginning their thirty years of service with trade publications. From Philadelphia he went to New York on the editorial staff on the *Metal Worker*, *Plumber*, and *Steam Fitter*.

In 1920, the Edwin A. Scott Publishing Co. was organized and the *Metal Worker*, *Plumber and Steam Fitter* was amalgamated with *Sheet Metal*. The two publications were subsequently reorganized as *Sheet Metal Worker and Sanitary and Heating Engineering*. Mr. Chew then became vice-president and a director in the publishing company and shared with Edwin A. Scott the editorial duties of *Sheet Metal Worker*. This position he continued to fill until the illness which resulted in his death set in.

Mr. Chew became a member of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in 1895. From the first he took an active part in the affairs of the Society. For three successive years, 1906-1908, he was elected a member of the Board of Governors and served on many committees. He was keenly interested in Chapter activities and was president of the New York Chapter in 1919.

He was an honorary member of the *National Warm Air Heating and Ventilating Association*, and the *National Association of Sheet Metal Contractors*, and a member of the *American Society of Sanitary Engineering*.

Not only was he a member of engineering and trade organizations, but he also belonged to several clubs and fraternal orders including the Eastern Trade Golf Association, the Newark Athletic Club, and the Knights Templar of the A. F. & A. M.

Mr. Chew is survived by an only son.

Members of the Society will miss his ready wit, his spectacular and quaintly original expressions and above all his wise counsel and loyal devotion in the work for the advancement of the profession and will mourn his loss as a loyal member and true friend of the organization.

John D. Cassell at the annual meeting held in January 1924 spoke of the illness of Mr. Chew whose genial and forceful personality was missing from the meeting. He offered the following resolution which was unanimously carried:

Resolved, that the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in Annual Meeting assembled hereby extend their sincere sympathy in his present affliction and hope for his speedy recovery; and that a copy of this Resolution be forwarded to Frank K. Chew.

President Addams in his opening address at the Semi-Annual Meeting paid tribute to Mr. Chew in the following words:

"The Society has felt the loss of Frank K. Chew whose untiring efforts and interest in the Society has been constant since 1895."

INDEX

	PAGE
ADDAMS, HOMER, Discussion of The Status of Domestic Oil Heating.....	237
Air Cleaners, Determining the Efficiency of, by A. M. GOODLOE.....	47
Air Dustiness, Production and Measurement of, by MARGARET INGELS.....	121
Air Handling and Humidity Problems in a Wisconsin Paper Mill, by ARTHUR T. NORTH.....	55
Air Leakage around Window Openings, by C. C. SCHRADER.....	313
Air Leakage through Openings in Buildings, by F. C. HOUGHTEN and C. C. SCHRADER.....	105
Discussion of.....	120
Air Motion—High Temperatures and Various Humidities—Reactions on Human Beings, by W. J. McCONNELL, F. C. HOUGHTEN and C. P. YAGLOGLOU.....	167
Air, Simultaneous Flow of, and Water in Pipes, by L. S. O'BANNON.....	157
Air Velocities, Cooling Effects on Human Beings Produced by Various, by F. C. HOUGHTEN and C. P. YAGLOGLOU.....	193
ANDERSON, F. P., Discussion of Measuring Heat Transmission in Building Structures and a Heat Transmission Meter.....	104
Annual Meeting, The Thirtieth.....	1
Anthracite and Bituminous Coal, Performance of a Warm Air Furnace with, by A. P. KRATZ.....	277
Application of the Heat Flow Meter, Practical, by P. NICHOLLS.....	289
BALDWIN, WILLIAM J., In Memoriam.....	404
BALLARD, A. H., The Status of Domestic Oil Heating.....	227
Discussion of The Status of Domestic Oil Heating.....	229, 230, 232
BARTLEY, F. C., Discussion of The Status of Domestic Oil Heating.....	232
Beings, Human, Cooling Effect Produced by Various Air Velocities, by F. C. HOUGHTEN and C. P. YAGLOGLOU.....	193
Beings, Human, Reactions on of Air Motion—High Temperatures and Various Humidities, by W. J. McCONNELL, F. C. HOUGHTEN and C. P. YAGLOGLOU.....	167
Bituminous and Anthracite Coal, Performance of a Warm Air Furnace with, by A. P. KRATZ.....	277
Body, Heat Given Up by the Human and Its Effect on Heating and Ventilation Problems, by C. P. YAGLOGLOU.....	365
Boilers, Code for Testing Low-Pressure Steam Heating.....	9
BOYD, D. KNICKERBACKER, Discussion of Cooling Effect on Human Beings Produced by Various Air Velocities.....	210

BROOKS, H. W., Discussion of Cooling Effect on Human Beings Produced by Various Air Velocities.....	211
Discussion of Determining the Efficiency of Air Cleaners.....	53
Discussion of An Improved Method of Determining the Heat Transfer through Wall, Floor, and Roof Sections.....	46
Discussion of The Status of Domestic Oil Heating.....	231
Buildings, Air Leakage through the Openings in, by F. C. HOUGHTEN and C. C. SCHRADER.....	105
Building Structures, Measuring Heat Transmission in and a Heat Transmission Meter, by P. NICHOLLS.....	65
By-Products in Flour Mill Heating and Humidifying, Using, by E. K. CAMPBELL.....	243
CALVERT, N. W., The Economical Utilization of Heat from Central Station Plants.....	21
Discussion of The Economical Utilization of Heat from Central Station Plants.....	39
CAMPBELL, E. K., Using By-Products in Flour Mill Heating and Humidifying...	243
CARRIER, W. H., Discussion of Air Leakage through the Openings in Buildings..	120
Discussion of Cooling Effect on Human Beings Produced by Various Air Velocities.....	209
Discussion of Determining the Efficiency of Air Cleaners.....	52
Discussion of Measuring Heat Transmission in Building Structures and a Heat Transmission Meter.....	104
Discussion of Problems in Ventilation of Department Stores.....	226
Central Station Plants, Economical Utilization of Heat from, by N. W. CALVERT and J. E. SEITER.....	21
CHEW, FRANK K., In Memoriam.....	405
CLARKSON, W. B., Discussion of An Improved Method of Determining the Heat Transfer through Wall, Floor and Roof Sections.....	46
Cleaners, Determining the Efficiency of Air, by A. M. GOODLOE.....	47
Coal, Performance of a Warm Air Furnace with Anthracite and Bituminous, by A. P. KRATZ.....	277
Code for Testing Low-Pressure Steam Heating Boilers.....	9
Comments on Present Day Heating and Ventilating Practice, by W. S. TIMMIS...	395
Condensate, Critical Velocity of Steam and Mixtures in Horizontal, Vertical and Inclined Pipes, by F. C. HOUGHTEN, LOUIS EBIN and R. L. LINCOLN..	139
Condensation, Flow of and Steam, as Affected by High Pressures, Horizontal Off-sets and Valves, by LOUIS EBIN and R. L. LINCOLN.....	323
Cooling Effect on Human Beings Produced by Various Air Velocities, by F. C. HOUGHTEN and C. P. YAGLOGLOU.....	193
Discussion of.....	209
Correlation of Skin Temperatures and Physiological Reactions, by W. J. MCCONNELL and C. P. YAGLOGLOU.....	305
Critical Velocity of Steam and Condensate Mixtures in Horizontal, Vertical and Inclined Pipes, by F. C. HOUGHTEN, LOUIS EBIN and R. L. LINCOLN..	139
Discussion of.....	155
DAY, V. S., Selecting Wall Stacks Scientifically for Gravity Warm Air Heating Systems.....	284
Department Stores, Problems in the Ventilation of, by A. M. FELDMAN.....	221

Determining Dry Return Proportions, by R. V. FROST.....	253
Determining the Efficiency of Air Cleaners, by A. M. GOODLOE.....	47
Discussion of.....	52
Determining the Heat Transfer through Wall, Floor and Roof Sections, An Improved Method of, by R. F. NORRIS, H. H. GERMOND and C. M. TUTTLE.....	41
Domestic Oil Heating, The Status of, by A. H. BALLARD.....	227
DONNELLY, J. A., Discussion of Critical Velocity of Steam and Condensate Mixtures in Pipes.....	156
Discussion of Simultaneous Flow of Water and Air in Pipes.....	166
Dry Return Proportions, Determining, by R. V. FROST.....	253
DUGAN, T. M., Discussion of Critical Velocity of Steam and Condensate Mixtures in Pipes.....	153
Dustiness, The Production and Measurement of Air, by MARGARET INGELS.....	121
EBIN LOUIS, Critical Velocity of Steam and Condensate Mixtures in Horizontal, Vertical and Inclined Pipes.....	139
Flow of Steam and Condensation as Affected by High Pressure, Horizontal Offsets and Valves.....	323
Economical Utilization of Heat from Central Station Plants, by N. W. CALVERT and J. E. SEITER.....	21
Discussion of.....	37
Effect on Heating and Ventilating Problems, Heat Given Up by the Human Body and Its, by C. P. YAGLOGLOU.....	365
Effect on Human Beings Produced by Various Air Velocities, Cooling, by F. C. HOUGHTEN and C. P. YAGLOGLOU.....	193
Effective Temperature Applied to Industrial Ventilation Problems, by C. P. YAGLOGLOU and W. E. MILLER.....	339
Effective Temperature Studies, Value of the Kata Thermometer in, by MARGARET INGELS.....	301
Efficiency of Air Cleaners, Determining the, by A. M. GOODLOE.....	47
EHRlich, M. W., Discussion of Economical Utilization of Central Station Heat.....	37, 39
Discussion of The Status of Domestic Oil Heating.....	230
EICHER, H. C., Discussion of The Status of Domestic Oil Heating.....	231
Electricity, The Place of in the General Heating Field, by LEE P. HYNES.....	213
Emission, Heat, from Heating Surfaces of Furnace, by A. P. KRATZ.....	59
FELDMAN, A. M., Problems in Ventilation of Department Stores.....	221
Discussion of The Place of Electricity in the General Heating Field....	220
Discussion of Problems in Ventilation of Department Stores.....	226
Field, General Heating, The Place of Electricity in, by LEE P. HYNES.....	213
Floor and Roof Sections, An Improved Method of Determining the Heat Transfer through Wall, by R. F. NORRIS, H. H. GERMOND and C. M. TUTTLE.....	41
Flour Mill Heating and Humidifying, Using By-Products in, by E. K. CAMPBELL.....	243
Flow of Steam and Condensation as Affected by High Pressures, Horizontal Offsets and Valves, by LOUIS EBIN and R. L. LINCOLN.....	323
Flow, Simultaneous of Water and Air in Pipes, by L. S. O'BANNON.....	157
FROST, R. V., Determining Dry Return Proportions.....	253
Discussion of Simultaneous Flow of Air and Water in Pipes.....	165
Furnace, Heat Emission from Heating Surfaces of, by A. P. KRATZ.....	59

Furnace, Warm Air, Performance of with Anthracite and Bituminous Coal, by A. P. KRATZ.....	277
GOODLOE, A. M., Determining the Efficiency of Air Cleaners.....	47
Discussion of Determining the Efficiency of Air Cleaners.....	54
Gravity Warm Air Heating Systems, Selecting Wall Stacks Scientifically for, by V. S. DAY.....	284
GERMOND, H. H., An Improved Method of Determining the Heat Transfer through Wall, Floor, and Roof Sections.....	41
HALLETT, E. S., Discussion of Air Leakage through Openings in Buildings.....	120
Discussion of Cooling Effect on Human Beings Produced by Various Air Velocities.....	155
Handling Air and Humidity Problems in a Wisconsin Paper Mill, by ARTHUR T. NORTH.....	55
HARDING, L. A., Discussion of Measuring Heat Transmission in Building Structures and a Heat Transmission Meter.....	102
HART, H. M., Discussion of Critical Velocity of Steam and Condensate Mixtures in Pipes.....	155, 156
Discussion of Economical Utilization of Central Station Heat.....	38, 39
Discussion of The Status of Domestic Oil Heating.....	231, 232
HARTMAN, F. E., Ozone and Its Uses in Ventilation.....	259
Heat Emission from Heating Surfaces of Furnace, by A. P. KRATZ.....	59
Discussion of.....	63
Heat Flow Meter, Practical Application of, by P. NICHOLLS.....	289
Heat From Central Station Plants, Economical Utilization of, by N. W. CALVERT and J. E. SEITER.....	21
Heat Given Up by the Human Body and Its Effect on Heating and Ventilating Problems, by C. P. YAGLOGLOU.....	365
Heat Transfer through Wall, Floor and Roof Sections, An Improved Method of Determining, by R. F. NORRIS, H. H. GERMOND and C. M. TUTTLE...	41
Heat Transmission in Building Structures and a Heat Transmission Meter, Measuring, by P. NICHOLLS.....	65
Heating and Humidifying, Using By-Products in Flour Mill, by E. K. CAMPBELL.....	243
Heating and Ventilating Practice, Some Comments on Present Day, by W. S. TIMMIS.....	395
Heating and Ventilating Problems, Heat Given Up by the Human Body and Its Effect on, by C. P. YAGLOGLOU.....	365
Heating Boilers, Code for Testing Low Pressure Steam.....	9
Heating Field, General, The Place of Electricity in, by LEE P. HYNES.....	213
Heating, The Status of Domestic Oil, by A. H. BALLARD.....	227
Heating Surfaces of Furnace, Heat Emission from, by A. P. KRATZ.....	59
Heating Systems, Selecting Wall Stacks Scientifically for Gravity Warm Air, by V. S. DAY.....	284
HECHLER, F. G., Discussion of Measuring Heat Transmission in Building Structures and a Heat Transmission Meter.....	102
HEDGES, H. B., Discussion of The Status of Domestic Oil Heating.....	232
HERTER, C. H., Discussion of Measuring Heat Transmission in Building Structures and a Heat Transmission Meter.....	103

High Pressures, Horizontal Offsets and Valves, Flow of Steam and Condensation as Affected by, by LOUIS EBIN and R. L. LINCOLN.....	323
High Temperatures and Various Humidities—Air Motion, Reactions on Human Beings, by W. J. McCONNELL, F. C. HOUGHTEN and C. P. YAGLOGLOU	167
HILL, E. VERNON, Discussion of The Cooling Effect on Human Beings Produced by Various Air Velocities.....	210
Discussion of Determining the Efficiency of Air Cleaners.....	54
Discussion of The Production and Measurement of Air Dustiness.....	138
Horizontal Offsets, Valves, and High Pressures, Flow of Steam and Condensation as Affected by, by LOUIS EBIN and R. L. LINCOLN.....	323
Horizontal, Vertical and Inclined Pipes, Critical Velocity of Steam and Condensate Mixtures in, by F. C. HOUGHTEN, LOUIS EBIN and R. L. LINCOLN.....	139
HOUGHTEN, F. C., Air Leakage through Openings in Buildings.....	105
Air Motion—High Temperatures and Various Humidities—Reactions on Human Beings.....	167
Cooling Effect on Human Beings Produced by Various Air Velocities..	193
Critical Velocity of Steam and Condensate Mixtures in Horizontal, Vertical and Inclined Pipes.....	139
Discussion of Cooling Effect on Human Beings Produced by Various Air Velocities.....	211
Discussion of Critical Velocity of Steam and Condensate Mixtures in Horizontal, Vertical and Inclined Pipes.....	156
HOWATT, JOHN, Discussion of Cooling Effect on Human Beings Produced by Various Air Velocities.....	210
Human Beings, Cooling Effect on Produced by Various Air Velocities, by F. C. HOUGHTEN and C. P. YAGLOGLOU.....	193
Human Beings, Reactions on of Air Motion—High Temperatures and Various Humidities, by W. J. McCONNELL, F. C. HOUGHTEN and C. P. YAGLOGLOU.....	167
Human Body, Heat Given Up by and Its Effect on Heating and Ventilating Problems, by C. P. YAGLOGLOU.....	365
Humidifying, Using By-Products in Flour Mill Heating and, by E. K. CAMPBELL.	243
Humidities, Various, and High Temperatures—Air Motion—Reactions on Human Beings, by W. J. McCONNELL, F. C. HOUGHTEN and C. P. YAGLOGLOU	167
Humidity Problems and Air Handling in a Wisconsin Paper Mill, by ARTHUR T. NORTH.....	55
HYNES, LEE P., The Place of Electricity in the General Heating Field.....	213
Improved Method of Determining the Heat Transfer through Wall, Floor and Roof Sections, by R. F. NORRIS, H. H. GERMOND and C. M. TUTTLE..	41
Discussion of.....	46
Inclined Pipes, Critical Velocity of Steam and Condensate Mixtures in Horizontal Vertical, and, by F. C. HOUGHTEN, LOUIS EBIN and R. L. LINCOLN..	139
Industrial Ventilating Problems, Effective Temperature Applied to, by C. P. YAGLOGLOU and W. E. MILLER.....	339
INGELS, MARGARET, The Production and Measurement of Air Dustiness.....	121
Value of the Kata Thermometer in Effective Temperature Studies.....	301
Discussion of Determining the Efficiency of Air Cleaners.....	52
Discussion of The Production and Measurement of Air Dustiness.....	138

INNIS, HELEN R., Discussion of The Economical Utilization of Central Station Heat.....	39
IRELAND, T. H., Discussion of Critical Velocity of Steam and Condensate Mixtures in Pipes.....	156
JELLETT, S. A., Discussion of The Place of Electricity in the General Heating Field.....	220
Discussion of Problems in Ventilation of Department Stores.....	226
JONES, W. T., Discussion of Economical Utilization of Central Station Heat.....	39
Kata Thermometer, Value of in Effective Temperature Studies, by MARGARET INGELS.....	301
KRATZ, A. P., Heat Emission from Heating Surfaces of Furnace.....	59
Performance of a Warm Air Furnace with Anthracite and Bituminous Coal.....	277
LANGENBERG, E. B., Discussion of Cooling Effect on Human Beings Produced by Various Air Velocities.....	211
Discussion of Heat Emission from Heating Surfaces of Furnace.....	64
Discussion of The Status of Domestic Oil Heating.....	231
Leakage, Air, around Window Openings, by C. C. SCHRADER.....	313
Leakage, Air, through Openings in Buildings, by F. C. HOUGHTEN and C. C. SCHRADER.....	105
LINCOLN, R. L., Critical Velocity of Steam and Condensate Mixtures in Horizontal, Vertical and Inclined Pipes.....	139
Flow of Steam and Condensation as Affected by High Pressures, Horizontal Offsets and Valves.....	323
Low-Pressure Steam Heating Boilers, Code for Testing.....	6
MCCOLL, J. R., Discussion of Cooling Effect on Human Beings Produced by Various Air Velocities.....	211
Discussion of Economical Utilization of Central Station Heat.....	38
MCCONNELL, W. J., Air Motion—High Temperatures and Various Humidities—Reactions on Human Beings.....	167
Correlation of Skin Temperatures and Physiological Reactions.....	305
MCINTIRE, J. F., Discussion of The Status of Domestic Oil Heating.....	232
Measurement of Air Dustiness, and the Production of, by MARGARET INGELS.....	121
Measuring Heat Transmission in Building Structures and a Heat Transmission Meter, by P. NICHOLLS.....	65
Discussion of.....	102
Meeting, The Thirtieth Annual.....	1
Meeting, The Semi-Annual, 1924.....	235
Memoriam, In.....	403
MENSING, F. D., Discussion of Cooling Effect on Human Beings Produced by Various Air Velocities.....	210
Method, An Improved of Determining the Heat Transfer through Wall, Floor and Roof Sections, by R. F. NORRIS, H. H. GERMOND and C. M. TUTTLE.....	41
Meter, Heat Flow, Practical Application of, by P. NICHOLLS.....	289

Meter, Heat Transmission, Measuring Heat Transmission in Building Structures and, by P. NICHOLLS.....	65
Mill, Flour, Using By-Products in Heating and Humidifying, by E. K. CAMPBELL.....	243
Mill, Paper, Air Handling and Humidity Problems in a Wisconsin, by ARTHUR T. NORTH.....	55
MILLER, W. E., Effective Temperature Applied to Industrial Ventilation Problems	339
Mixtures, Critical Velocity of Steam and Condensate, in Horizontal, Vertical and Inclined Pipes, by F. C. HOUGHTEN, LOUIS EBIN and R. L. LINCOLN.....	139
Modern Trend in the Science of Ventilation, by PERRY WEST.....	377
Motion, Air—High Temperatures and Various Humidities, Reaction s on Human Beings, by W. J. MCCONNELL, F. C. HOUGHTEN and C. P. YAGLOGLOU.....	167
NICHOLLS, P., Measuring Heat Transmission in Building Structures and a Heat Transmission Meter.....	65
Practical Applications of the Heat Flow Meter.....	289
Discussion of Determining the Efficiency of Air Cleaners.....	53
Discussion of An Improved Method of Determining the Heat Transfer through Wall, Floor and Roof Sections.....	46
Discussion of Measuring Heat Transmission in Building Structures and a Heat Transmission Meter.....	104
NICHOLS, G. B., Discussion of Economical Utilization of Central Station Heat...	38
NORRIS, R. F., An Improved Method of Determining the Heat Transfer through Wall, Floor and Roof Sections.....	41
NORTH, ARTHUR T., Air Handling and Humidity Problems in a Wisconsin Paper Mill.....	55
O'BANNON, L. S., Simultaneous Flow of Water and Air in Pipes.....	157
Discussion of Simultaneous Flow of Water and Air in Pipes.....	166
Offsets, Horizontal and Valves, Flow of Steam and Condensation as Affected by High Pressures, by LOUIS EBIN and R. L. LINCOLN.....	323
Oil Heating, The Status of Domestic, by A. H. BALLARD.....	227
Openings in Buildings, Air Leakage through, by F. C. HOUGHTEN and C. C. SCHRADER.....	105
Openings, Window, Air Leakage around, by C. C. SCHRADER.....	313
Ozone and Its Use in Ventilation, by FRANK E. HARTMAN.....	259
Paper Mill, Air Handling and Humidity Problems in a Wisconsin, by ARTHUR T. NORTH.....	55
Performance of a Warm Air Furnace with Anthracite and Bituminous Coal, by A. P. KRATZ.....	277
Physiological Reactions, Correlation of Skin Temperature and, by W. J. MCCONNELL and C. P. YAGLOGLOU.....	305
Pipes, Horizontal, Vertical and Inclined, Critical Velocity of Steam in, by F. C. HOUGHTEN, LOUIS EBIN and R. L. LINCOLN.....	139
Pipes, Simultaneous Flow of Water and Air in, by L. S. O'BANNON.....	157
Place of Electricity in the General Heating Field by LEE P. HYNES.....	213
Discussion of.....	220

Plants, Central Station, Utilization of Heat From, by N. W. CALVERT and J. E. SEITER.....	21
Practical Application of the Heat Flow Meter, by P. NICHOLLS.....	289
Practice, Some Comments on Present Day Heating and Ventilating, by W. S. TIMMIS.....	395
PRATT, E. D. Discussion of Determining the Efficiency of Air Cleaners.....	52
Present Day Heating and Ventilating Practice, Some Comments on, by W. S. TIMMIS.....	395
Pressures, High, Horizontal Offsets and Valves, Flow of Steam and Condensation as Affected by, by LOUIS EBIN and R. L. LINCOLN.....	323
Problems, Air Handling and Humidity, in a Wisconsin Paper Mill, by ARTHUR T. NORTH.....	55
Problems, Heating and Ventilating, Heat Given Up by the Human Body and Its Effect on, by C. P. YAGLOGLOU.....	365
Problems in the Ventilation of Department Stores, by A. M. FELDMAN.....	221
Discussion of.....	226
Problems, Industrial Ventilation, Effective Temperature Applied to, by C. P. YAGLOGLOU and W. E. MILLER.....	339
Production and Measurement of Air Dustiness, by MARGARET INGELS.....	121
Discussion of.....	138
Proportions, Determining Dry Return, by R. V. FROST.....	253
Reactions on Human Beings, Air Motion—High Temperatures and Various Humidities, by W. J. McCONNELL, F. C. HOUGHTEN and C. P. YAGLOGLOU.....	167
Reactions, Physiological, Correlation of Skin Temperatures and, by W. J. McCONNELL and C. P. YAGLOGLOU.....	305
Return Proportions, Determining Dry, by R. V. FROST.....	253
RICHARDSON, D. R., Discussion of Heat Emission from Heating Surfaces of Furnace Roof Sections, An Improved Method of Determining the Heat Transfer through Wall, Floor, and, by R. F. NORRIS, H. H. GERMOND and C. M. TUTTLE.....	41
SCHRADER, C. C., Air Leakage around Window Openings.....	313
Air Leakage through the Openings in Buildings.....	105
Science of Ventilation, Modern Trend in the, by PERRY WEST.....	377
Sections, Wall, Floor, and Roof, An Improved Method of Determining the Heat Transfer through, by R. F. NORRIS, H. H. GERMOND and C. M. TUTTLE.....	41
SEITER, J. E., The Economical Utilization of Heat from Central Station Plants.....	21
Selecting Wall Stacks Scientifically for Gravity Warm Air Heating Systems, by V. S. DAY.....	284
Semi-Annual Meeting, The 1924.....	235
SEWELL, J. M., Discussion of The Status of Domestic Oil Heating.....	229
Simultaneous Flow of Water and Air in Pipes, by L. S. O'BANNON.....	157
Discussion of.....	165
Skin Temperatures and Physiological Reactions, Correlation of, by W. J. McCONNELL and C. P. YAGLOGLOU.....	305
Some Comments on Present Day Heating and Ventilating Practice, by W. S. TIMMIS.....	395
Stacks, Wall, Selecting Scientifically for Gravity Warm Air Heating Systems, by V. S. DAY.....	284

Station Plants, Central, Economical Utilization of Heat from, by N. W. CALVERT and J. E. SEITER.....	21
Status of Domestic Oil Heating, The, by A. H. BALLARD.....	227
Discussion of.....	229
Steam and Condensate Mixtures in Horizontal, Vertical, and Inclined Pipes, Critical Velocity of, by F. C. HOUGHTEN, LOUIS EBIN and R. L. LINCOLN.....	139
Steam and Condensation, Flow of as Affected by High Pressures, Horizontal Offsets and Valves, by LOUIS EBIN and R. L. LINCOLN.....	323
Steam Boilers, Code for Testing Low-Pressure.....	9
STILL, F. R., Discussion of Cooling Effect on Human Beings Produced by Various Air Velocities.....	211
Discussion of Determining the Efficiency of Air Cleaners.....	53
Stores, Problems in the Ventilation of Department, by A. M. FELDMAN.....	221
Structures, Measuring Heat in Building, and a Heat Transmission Meter, by P. NICHOLLS.....	65
Studies, Effective Temperature, The Value of the Kata Thermometer in, by MARGARET INGELS.....	301
Surfaces of Furnace, Heat Emission from Heating, by A. P. KRATZ.....	59
Systems, Gravity Warm Air Heating, Selecting Wall Stacks Scientifically for, by V. S. DAY.....	284
TAGGART, R. C., Discussion of Critical Velocity of Steam and Condensate Mixtures in Pipes.....	153
Temperature, Effective Applied to Industrial Ventilation Problems, by C. P. YAGLOGLOU and W. E. MILLER.....	339
Temperature, Effective Studies, Value of the Kata Thermometer in, by MARGARET INGELS.....	301
Temperatures, High and Various Humidities—Air Motion—Reactions on Human Beings, by W. J. MCCONNELL, F. C. HOUGHTEN and C. P. YAGLOGLOU.....	167
Temperatures, Skin and Physiological Reactions, Correlation of, by W. J. MCCONNELL and C. P. YAGLOGLOU.....	305
Testing Low Pressure Steam Boilers, Code for.....	9
Thermometer, Kata, Value of in Effective Temperature Studies, by MARGARET INGELS.....	301
TIMMIS, W. S., Some Comments on Present Day Heating and Ventilating Practice.....	395
Transfer, Heat, An Improved Method of Determining through Wall, Floor and Roof Sections, by R. F. NORRIS, H. H. GERMOND and C. M. TUTTLE.....	41
Transmission Heat, Measuring in Building Structures and a Heat Transmission Meter, by P. NICHOLLS.....	65
Trend, Modern in the Science of Ventilation, by PERRY WEST.....	377
TUTTLE, C. M., An Improved Method of Determining Heat Transfer through Wall, Floor and Roof Sections.....	41
Use in Ventilation, Ozone and Its, by F. E. HARTMAN.....	259
Using By-Products in Flour Mill Heating and Humidifying, by E. K. CAMPBELL.....	243
Utilization of Heat from Central Station Plants, Economical by N. W. CALVERT and J. E. SEITER.....	21
Value of the Kata Thermometer in Effective Temperature Studies, by MARGARET INGELS.....	305

Valves, Flow of Steam and Condensation as Affected by High Pressures, Horizontal Offsets and, by LOUIS EBIN and R. L. LINCOLN.....	323
VAN DEUSEN, M. S., Discussion of Measuring Heat Transmission in Building Structures and a Heat Transmission Meter.....	103
Discussion of An Improved Method of Determining the Heat Transfer through Wall, Floor and Roof Sections.....	46
Various Air Velocities, Cooling Effect on Human Beings Produced by, by F. C. HOUGHTEN and C. P. YAGLOGLOU.....	193
Various Humidities, Air Motion, High Temperatures—Reactions on Human Beings, by W. J. McCONNELL, F. C. HOUGHTEN and C. P. YAGLOGLOU.....	167
Velocities, Various Air, Cooling Effect on Human Beings Produced by, by F. C. HOUGHTEN and C. P. YAGLOGLOU.....	193
Velocity Critical, of Steam and Condensate Mixtures in Horizontal, Vertical and Inclined Pipes by F. C. HOUGHTEN, LOUIS EBIN and R. L. LINCOLN....	139
Ventilating Practice, Some Comments on Present Day Heating and, by W. S. TIMMIS.....	395
Ventilating Problems, Heat Given Up by the Human Body and Its Effect on Heating and, by C. P. YAGLOGLOU.....	365
Ventilation of Department Stores, Problems in, by A. M. FELDMAN.....	221
Ventilation, Ozone and Its Use in, by FRANK E. HARTMAN.....	259
Ventilation, Modern Trend in the Science of, PERRY WEST.....	377
Ventilation Problems, Effective Temperature Applied to Industrial, by C. P. YAGLOGLOU and W. E. MILLER.....	339
Vertical, Horizontal and Inclined Pipes, Critical Velocity of Steam and Condensate Mixtures in, by F. C. HOUGHTEN, LOUIS EBIN and R. L. LINCOLN....	139
Wall, Floor and Roof Sections, An Improved Method of Determining the Heat Transfer through, by R. F. NORRIS, H. H. GERMOND and C. M. TUTTLE.....	41
Wall Stacks, Selecting Scientifically for Gravity Warm Air Heating Systems, by V. S. DAY.....	284
Warm Air Furnace, Performance of with Anthracite and Bituminous Coal, by A. P. KRATZ.....	277
Warm Air Heating Systems, Selecting Wall Stacks Scientifically for Gravity, by V. S. DAY.....	284
Water and Air, Simultaneous Flow of in Pipes, by L. S. O'BANNON.....	157
WECHSLER, G. A., Discussion of The Status of Domestic Oil Heating.....	231
WEST, PERRY, Modern Trend in the Science of Ventilation.....	377
Window Openings, Air Leakage around, by C. C. SCHRADER.....	313
Wisconsin Paper Mill, Air Handling and Humidity Problems in, by ARTHUR T. NORTH.....	55
WOOLSTON, IRA, Discussion of The Status of Domestic Oil Heating.....	232
YAGLOGLOU, C. P., Air Motion—High Temperatures and Various Humidities—Reactions on Human Beings.....	167
Cooling Effect on Human Beings Produced by Various Air Velocities..	193
Correlation of Skin Temperatures and Physiological Reactions.....	305
Effective Temperature Applied to Industrial Ventilation Problems...	339
Heat Given Up by the Human Body and Its Effect on Heating and Ventilating Problems.....	365

